VOLUME I

THERMODYNAMICS

BY

H. KERR THOMAS, M.I.MECH.E., M.I.A.E.

FUEL TECHNOLOGY

BY

H. KERR THOMAS, M.I.MECH.E., M.I.A.E.

PETROL ENGINE THEORY

BY

A. W. JUDGE, A.R.C.Sc., WH.Sc., A.M.I.A.E.

CYLINDERS, CYLINDER HEADS, AND LINERS

BY

S. W. NIXON, M.Sc., A.M.I.A.E.

MECHANICS OF A MOVING VEHICLE

BY

H. KERR THOMAS, M.I.MECH.E., M.I.A.E.

AUTOMOBILE ENGINEERING

A PRACTICAL AND AUTHORITATIVE WORK FOR AUTOMOBILE ENGINEERS, DESIGNERS AND STUDENTS

EDITED BY

H. KERR THOMAS

M.I.MECH.E., M.I.A.E.

VOLUME I

SECOND EDITION

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AUTOMOBILE ENGINEERING

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PREFACE

An excellent reception was accorded to the first edition of *Automobile Engineering* by engineers and engineering students, and in consequence the Publishers have not deemed it advisable to depart to a great extent from the form or substance of the first edition. As hitherto, Volume I is mainly devoted to a concise and clear explanation of general principles, or in other words theory. All five sections have been carefully checked over by the various writers and brought completely up to date. New illustrations have been included where necessary and some important additions have been made to the text.

In the section on Fuel Technology mention is made of recent developments in the production of anti-detonating fuels for high efficiency engines; in the section on Petrol Engine Theory new data on compression ratios, aluminium cylinder heads, etc., has been incorporated, and in the section on Cylinders, Cylinder Heads, and Liners, much additional information is to be found. Reference is made to crankcase stiffening, copper alloy cylinder heads, cylinder distortion, combustion chamber formation, cylinder wear, etc. The notes on the latest austenitic cylinder liners should be of special interest in view of their long life. Of equal importance is the survey of the causes of cylinder wear, confirmed by recent research into this most important problem.

PREFACE
TO FIRST EDITION

THIS first volume of *Automobile Engineering* contains five sections which, as will be seen, are entirely devoted to theory. It would be useless to commence the study of Automobile Engineering without a general understanding of thermodynamics. In the first section this subject is sufficiently covered to enable the student to avoid many pitfalls; for the more advanced reader, the section supplies formulae for most of the problems he is likely to meet.

The section on Fuel Technology presents the general aspects of this branch of the work as briefly as possible, and provides such information as is commonly needed by designers.

The third section covers the general theory of petrol engines, and develops the work of the second section.

Section V, which completes this volume, deals with the general applied mechanics of a vehicle on the road under actual running conditions, enabling the student, at an early stage, to become familiar with the correlation of theory and practice in his work, and providing him with information which will be required in the actual designing of a motor vehicle.

H. K. T.

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SECTION I

THERMODYNAMICS

BY

H. KERR THOMAS, M.I.MECH.E., M.I.A.E.

SECTION
THERMODYNAMICS

INTRODUCTORY

In order to arrive at a clear understanding of what takes place in the cylinder of an internal combustion engine, it is necessary to consider the general subject of thermodynamics, or the relationship between heat and mechanical work; and to narrow this down to the limits of this section, we may state at the outset that all internal combustion engines are akin to the hot air engine, and for the most part, as will be seen as we proceed, the investigations on internal combustion engines are based on an assumption that pure dry air is the working substance in the cylinder. In fact, at the commencement of each cycle of operations this is very nearly true. This air is heated to a very high temperature by the combustion of liquid or partially vaporized fuel within the working cylinder, after which a change takes place through the combination of the oxygen of the air with the hydrogen and carbon constituents of the fuel, so that the end of each working cycle is accomplished with a working substance consisting mainly of a mixture of nitrogen, water vapour, and carbonic acid, with traces of other substances, depending on circumstances. The study of these processes with the resulting changes of pressure, volume, and temperature is our immediate task.

We are now concerned with the measurement of the energy contained in the working substance, and although it is not possible by any method at present known to measure the whole amount of energy it

contains, it is possible to measure *changes* in that amount, changes due to the substance receiving or rejecting heat. The first thing to remember is that to do work requires the expenditure of heat, and conversely to produce heat necessitates the performance of work, so that when a substance receives heat while not working, its stock of energy is increased by an amount equivalent to the heat received; and if the

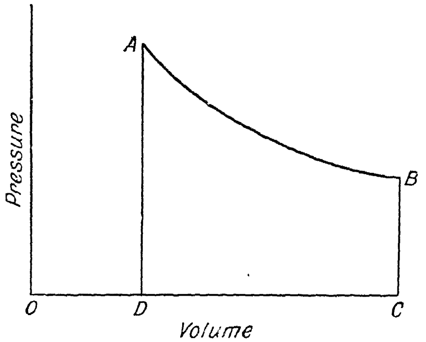


FIG. 1

substance is made to perform work without taking in heat, such work can only be done by drawing on the stock of energy in the substance, which will lose a quantity of heat equivalent to the amount of work done.

These changes of condition are usually represented by what is known as a pressure volume diagram, shown by Fig. 1, and from this, as will be explained later, the changes in the energy content of the substance can be measured.

In an engine of the usual type, with a cylinder and piston, the working substance performs work by the agency of change in its volume, and the quantity of

work done depends on the relation of the pressure to the volume which occurs during the change. If we refer to Fig. 1 we understand it represents a change of volume from OD to OC, with a change of pressure from DA to CB. Starting at this point A, the pressure is DA and the volume OD, we now imagine the working substance to expand to a volume OC, we shall find that the pressure has dropped to CB, and the curve AB will

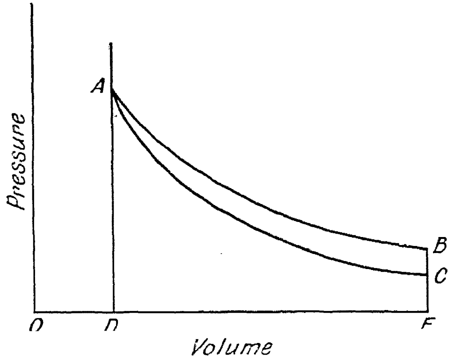


FIG. 2

represent all the conditions of volume and pressure. Then the work done by the substance during expansion may be represented by the area $ABCD$ beneath the curve AB.

Again, let us consider an operation in which the substance is not only expanded but compressed during the operation, thus passing through a complete cycle as shown in Fig. 2. We start at the point C and compress the substance from OE to OD, we shall find its pressure has risen to some point A, if it is next allowed to perform work by expanding from volume D to volume E its pressure will fall from, say, A to B. Then, as in the former case, the work done during expansion is

represented by the area of the figure *ABED*, while the work performed on the substance during compression is represented by the area *ACED*, and the net amount of work we shall obtain is the difference of these two, or *ABED* - *ACED*, or in other words, the net work given out by the substance during the cycle, is equal to the area of the closed figure *ABCA* representing the complete cycle. Such a diagram as Fig. 2 is, of course, the ordinary indicator diagram which we can obtain from any kind of gas-pressure engine which has a cylinder and a piston.

In the action of any heat engine the working substance returns periodically to the same conditions of pressure, temperature, and volume, so that we speak of its having passed through a "cycle," and for every such cycle heat is taken in, part of it is converted into work, and the remainder is discharged unused, and for the whole process a definite relationship or equation must hold good, viz.: Heat supplied = heat discharged + heat equivalent of work done.

Laws of Thermodynamics. There are two laws on which the whole system of thermodynamics is built up. Ewing has stated them as follows—

1. When mechanical energy is produced from heat, a definite quantity of heat goes out of existence for every unit of work done; and conversely, when heat is produced by the expenditure of mechanical energy, the same quantity of heat comes into existence for every unit of work expended.

2. It is impossible for a self-acting machine, unaided by any external agency, to convey heat from one body to another at a higher temperature.

We may express this in another way by saying that heat cannot pass from a colder to a hotter body.

Units of Force, Pressure, and Work. Since we have adopted the pound as the standard of weight, the

standard of pressure becomes the pound per square inch, and we may observe here that the pressure of the atmosphere at sea level at the latitude of London is 14·689 lb. per sq. in.; and since small pressures are sometimes measured in inches of mercury, we may add that 1 in. of mercury (at freezing point) is equivalent to 0·4912 lb. per sq. in.

The unit of work is the foot-pound, and the unit of heat is the British Thermal Unit or B.Th.U., the amount of heat required to raise the temperature of 1 lb. of water 1° F. The relationship between the performance of mechanical work and the expenditure of heat, first established by Joule, is a fixed quantity, known as the Mechanical Equivalent of Heat, or "Joule's Equivalent." It was ascertained by agitating a known quantity of water in an insulated vessel by means of a paddle, caused to revolve by means of a cord attached to a falling weight, and observing the rise in the temperature of the water compared with the distance fallen by the weight in the same time. Joule's results have been checked by other investigators, and the ratio is now accepted as 777·8 ft.-lb. (usually taken as 778 ft.-lb), being the mechanical equivalent of one B.Th.U.

Properties of Gases. In discussing thermodynamics it is usual to assume the existence of a "perfect gas" which exactly conforms to certain conditions we are about to describe, and which are only very nearly true of real gases. Apparently hydrogen most nearly approaches the defined properties of a perfect gas, but as its liquefying point is approached, its behaviour grows more erratic.

There is a certain code of behaviour for all gases which is defined by three laws, called after their respective discoverers the laws of Boyle, Charles, and Joule. Boyle's law may be explained thus: If we

imagine a cylinder fitted with a frictionless piston (see Fig. 3) and with some means of maintaining the temperature constant, as the piston is forced down, the volume of air will decrease, and the pressure will increase in inverse ratio, so that the pressure P multiplied by the volume V is always a constant quantity.

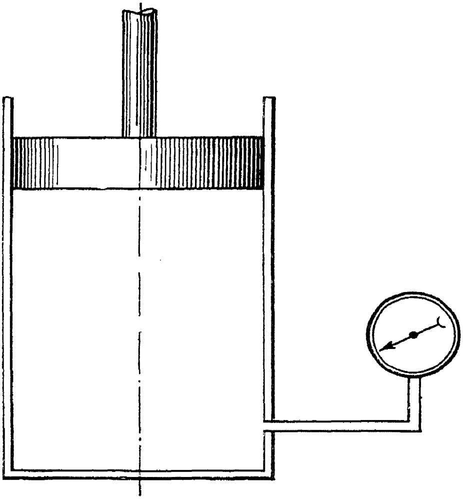


FIG. 3

Whether the piston be moved in or out, the volume of the air will always be inversely proportional to its pressure, provided its temperature remains constant. When the pressure is doubled the volume is halved, and *vice versa*, and no matter what the pressure and volume may be, their product is always the same. Hence Boyle's law may be stated thus: "At constant temperature, the volume of a gas varies inversely as its pressure."

The law can be expressed algebraically thus

$$= \text{a constant.}$$

Where P = pressure, and V = volume.

Boyle's law is very nearly, but not exactly, true.

Charles' law may be explained by assuming the same piston and cylinder (Fig. 3) in which the air is at a known pressure and volume, P and V; we also know its temperature T, which we assume we have some means of varying. We increase the temperature by a known amount, x, then its new temperature has become $T + x$. One of two things must happen: (*a*) if we prevent the piston from moving, i.e. we maintain a constant volume, the pressure will rise by an amount we will call y and will become $P + y$. (*b*) If we allow the piston to move freely and so maintain a constant pressure, the volume will increase by an amount we will call z, and will become $V + z$. Experiment proves the very important fact that if the temperature is increased by $2x$ the volume will increase by $2z$, and so on, and we can state Charles' law thus: "At constant pressure, the volume of a gas increases by equal amounts for equal increments of temperature."

The law can be expressed algebraically, thus—

If P, T, and V be initial pressures, temperatures, and volumes, and P_1, T_1, and V_1 be final values of the same, then

$$\frac{P}{P_1} = \frac{T}{T_1} \text{ or } \frac{V}{V_1} = \frac{T}{T_1}$$

showing that the ratio of pressure or volume varies as the ratio of temperature. We shall shortly reach a very important conclusion from this law.

Joule's law deals with the internal energy of a gas, and is of a somewhat more abstract nature. Joule found from experiment that if a gas is allowed to

expand without doing any external work, while its pressure falls, its temperature does not change and consequently its stock of internal energy does not change either. Hence the law may be stated thus: "The internal energy of a gas depends only on its temperature."

These three Laws, in the hands of later investigators equipped with improved apparatus, have been shown to be only approximately true for any actual gas, since what has been considered as a perfect gas does not actually exist.

Absolute Temperature. What is known as absolute temperature is of fundamental importance in all problems of thermodynamics. We have seen from Charles' law that at constant pressure the volume of a gas increases or decreases by a fixed amount for equal changes of temperature. It has been established that at a temperature of freezing point, or 0° C., if the pressure be maintained constant, a rise of 1° C. will increase the volume of a gas by $\frac{1}{273}$rd part. Similarly, if it be cooled by 1° C. the volume will decrease by $\frac{1}{273}$rd part. It follows that if the gas, at a temperature of 0° C., were cooled by 273° C. its volume would be zero and the temperature would be − 273° C. This point is therefore known as the absolute zero of temperature, and may be assumed to be the point at which "heat" ceases to exist.

This conception of an absolute zero of temperature leads to the further one, that at that point all gases would cease to exist, but fortunately our imagination is not taxed quite so far, since every known gas condenses to a liquid at an appreciable temperature above absolute zero. By what is known as the cascade system, however, where a liquefied gas is allowed to evaporate after being cooled by the evaporation of another, temperatures have been reached within $\frac{1}{4}$ degree of absolute zero.

To convert the Centigrade temperature into Fahrenheit, which we have adopted as standard—

$$(\text{C.}^\circ \times \tfrac{9}{5}) + 32^\circ = \text{F.}^\circ$$

So in this case

$$\text{absolute zero F.} = -(273 \times \tfrac{9}{5}) + 32$$
$$0^\circ \text{ abs. F.} = -491 + 32$$
$$0 \text{ abs. F.} = -459^\circ$$

Specific Heat of Gases. The specific heat of a gas, like any other substance, is the *relative* quantity of heat required to raise a unit weight of the gas by a fixed amount (say, 1° F.), but in dealing with gases we have to distinguish between two different methods of heating. The gas may be heated under a condition of constant volume or constant pressure, the quantities of heat required under the two conditions not being the same. We use the symbol K_p to represent the specific heat at constant pressure, and K_v for the specific heat at constant volume.

We will first consider the process of heating a unit quantity of a perfect gas at constant volume, from absolute temperature T to absolute temperature T_1. Then the heat absorbed by the gas is $K_v(T_1 - T)$. Note that the volume does not change, and therefore no external work is done; all this heat has gone to increase the stock of internal energy of the gas. We next consider the other method of heating, viz. at constant pressure. In this case the heat taken is $K_p(T_1 - T)$. But the gas expands, and therefore external work is done, its amount being $P(V_1 - V)$, where V_1 and V are the volume at the end and beginning of the process respectively, and P is the pressure which is constant.

Expansion and Compression of Gases. There are two kinds of reversible expansion or compression: (1) Isothermal and (2) Adiabatic. Isothermal expansion or

compression means that during expansion heat is received from an outside source, and during compression heat is rejected to an outside source, sufficient in each case to maintain the temperature constant throughout the process. Adiabatic expansion or compression means that during either process no heat enters the substance from outside sources, or conversely, leaves it.

In actual engines neither action is strictly adiabatic, as there is always some interchange of heat between the cylinder and piston and the working substance, but at the high speed of modern engines the time element is so short that the interchange is negligible.

During isothermal expansion the gas must receive heat from external sources to maintain its temperature constant, and to the same end it must, during isothermal compression, be in contact with some receiver which can absorb the heat generated. Generally speaking, we are not concerned with isothermal compression or expansion in calculations for internal combustion engines. During isothermal compression or expansion the expression PV equals a constant, since the pressure varies inversely as the volume, and accordingly any isothermal line on the pressure volume diagram is a rectangular hyperbola.

So if P and V represent the absolute initial pressure and volume, and P_1 and V_1 the final pressure and volume, then

$$P_1V_1 = PV$$

For example, let $P = 20$ lb. per sq in. abs.
and $V = 200$ cub. in.
then $PV = 20 \times 200 = 4000$

suppose the gas compressed isothermally to a volume of 100 cub. in.

then $P_1 \times 100 = PV = 4000$
so $P_1 = 4000/100 = 40$

Hence the volume of 200 cub. in. at 20 lb. per sq. in abs. has become 100 cub. in. at 40 lb. per sq. in. abs.

It will be as well to define here precisely what is understood by compression ratio. The volume of a gas which is operated on in the cylinder at each cycle consists of the working volume swept through by the piston at each stroke, plus the clearance space into which the gas is compressed.

Thus the total working volume = swept volume plus clearance = V, while the final volume = clearance volume = V_1.

Hence the compression ratio

$$= \frac{\text{Swept volume} + \text{clearance volume}}{\text{Clearance volume}} = \frac{V}{V_1}$$

During adiabatic compression or expansion, as we have seen, owing to the fact that the temperature of the gas varies, the results are modified by the specific heat of the gas, i.e. the amount of heat (and therefore internal energy) it can absorb. We have already discussed the difference between the specific heats of a gas at constant temperature and constant volume, and we express the ratio between the two as specific heat at constant pressure/specific heat at constant volume, for which we employ the symbol γ. It can then be shown that

$$PV^{\gamma} = \text{a constant}$$

and

$$TV^{\gamma\text{-}1} = \text{a constant}$$

The specific heat of air at constant pressure and at 60° F. is 0·2375, and at constant volume it is 0·1691, so we have the ratio

$$\frac{K_p}{K_v} = \frac{0{\cdot}2375}{0{\cdot}1691} = 1{\cdot}405$$

$$\text{hence } \gamma = 1{\cdot}4 \text{ and } \gamma - 1 = 1{\cdot}4 - 1 = 0{\cdot}4$$

It is important to appreciate clearly the difference between isothermal and adiabatic compression and expansion as it affects the ultimate results. Isothermal curves are very simple to calculate, and assuming a compression ratio of 5 to 1, and commencing with compression, we have an initial absolute pressure of 1 atmosphere = 14·7 lb. per sq. in., and 5 volumes of pure air in the cylinder; we will assume the volume is successively reduced to 4, 3, 2, and 1, allowing sufficient time in the process for the disappearance of the heat normally produced by compression. Then at the commencement we have $V = 5$ and $P = 1$, so $PV = 5$.

At 4 volumes PV still equals 5, but V is only 4,

so at 4 vols. $P = 5/4$ atmospheres
,, 3 ,, $P = 5/3$,,
,, 2 ,, $P = 5/2$,,
,, 1 ,, $P = 5/1$,,

multiplying each of these fractions by 14·7 we obtain the absolute pressures at each point of compression, viz.—

5 vols.	$P = 14{\cdot}7$	lb. sq. in. abs.	$5 \times 14{\cdot}7 = 73{\cdot}5$	$= PV$
4 ,,	$P = 18{\cdot}37$	,,	$4 \times 18{\cdot}37 = 73{\cdot}5$	$= PV$
3 ,,	$P = 24{\cdot}50$	,,	$3 \times 24{\cdot}50 = 73{\cdot}5$	$= PV$
2 ,,	$P = 36{\cdot}75$	,,	$2 \times 36{\cdot}75 = 73{\cdot}5$	$= PV$
1 vol.	$P = 73{\cdot}50$	..	$1 \times 73{\cdot}50 = 73{\cdot}50$	$= PV$

We next calculate the adiabatic compression pressures starting from 5 volumes at 1 atmosphere, the equation being

$$PV^{\gamma} = \text{a constant, and } \gamma = 1{\cdot}4 \text{ for air;}$$

$$P_1 = P \times \left(\frac{V}{V_1}\right)^{1\cdot4}$$

For this we must make use of logarithms $P = 14{\cdot}7$.

At Volume	$\frac{V}{V_1}$	$\log \frac{V}{V_1}$	$\log \frac{V}{V_1} \times 1{\cdot}4$	$\left(\frac{V}{V_1}\right)^{1\cdot4} \times 14{\cdot}7$
				lb. sq. in. abs.
4	$5/4 = 1{\cdot}25$	·0969	$\cdot1356 = \log 1{\cdot}366$	20
3	$5/3 = 1{\cdot}66$	·2216	$\cdot3102 = \log 2{\cdot}042$	30
2	$5/2 = 2{\cdot}50$	·3979	$\cdot5570 = \log 3{\cdot}605$	53
1	$5/1 = 5{\cdot}00$	·6989	$\cdot9784 = \log 9{\cdot}515$	139·8

And for adiabatic expansion starting from 73·5 lb. at 1 volume.

1 vol.

2 vols. $P_1 = P_2 \times 2^{\gamma} = P_2 \times \log 2 \times 1{\cdot}4$;
$\log 2 = \cdot30103, \times 1{\cdot}4 = \cdot4214$

3 ,, $P_1 = P_3 \times 3^{\gamma} = P_3 \times \log 3 \times 1{\cdot}4$;
$\log 3 = \cdot47712, \times 1{\cdot}4 = \cdot6679$

4 ,, $P_1 = P_4 \times 4^{\gamma} = P_4 \times \log 4 \times 1{\cdot}4$;
$\log 4 = \cdot60206, \times 1{\cdot}4 = \cdot8428$

5 ,, $P_1 = P_5 \times 5^{\gamma} = P_5 \times \log 5 \times 1{\cdot}4$;
$\log 5 = \cdot69897, \times 1{\cdot}4 = \cdot9785$

$\cdot4214 = \log 2{\cdot}639$; $\cdot6679 = \log 4{\cdot}655$; $\cdot8428 = \log 6{\cdot}964$ and $\cdot9785 = \log 9{\cdot}517$ which are the values of $(V/V_1)^{\gamma}$

The equation is $P_1 = P_2 \times (V/V_1)^{\gamma}$ and $P_1 = 73{\cdot}5$, substituting the values of $(V/V_1)^{\gamma}$ above

$73{\cdot}5 = P_2 \times 2{\cdot}639$ so $P_2 = 73{\cdot}5/2{\cdot}639 = 27{\cdot}85$ lb. sq. in. abs.

$73{\cdot}5 = P_3 \times 4{\cdot}655$,, $P_3 = 73{\cdot}5/4{\cdot}655 = 15{\cdot}79$,,

$73{\cdot}5 = P_4 \times 6{\cdot}964$,, $P_4 = 73{\cdot}5/6{\cdot}964 = 10{\cdot}55$,,

$73{\cdot}5 = P_5 \times 9{\cdot}517$,, $P_5 = 73{\cdot}5/9{\cdot}517 = 7{\cdot}73$,,

In Fig. 4 these figures have been plotted and they show that isothermal compression of air to 5/1 compression ratio, will raise the pressure from that of the atmosphere to 73·5 lb. per sq. in., but if we do not allow any of the heat to escape (in adiabatic compression) the pressure will rise to 139·8 lb. per sq. in. Similarly, if we expand air from 73·5 lb. pressure isothermally to 5 times its volume, it will return to exactly atmospheric pressure (14·7 lb.), but if we expand it adiabatically from 73·5 lb. and do not maintain the temperature by supplying heat from an outside source, the pressure will fall to 7·73 lb. or approximately half an atmosphere. Note that adiabatic curves are steeper than isothermal.

Combustion. We have already noted that the products of combustion form part of the working substance, and this materially affects, as we shall see later, the value of γ. For purposes of comparison, however, following the investigations of a committee of the Institution of Civil Engineers, what is known as the "Air Standard" is taken as a measure of the ideal efficiency of an internal combustion engine. This assumes—

1. No transfer of heat between the working substance and the cylinder or piston.
2. Instantaneous complete combustion.
3. No change of specific volume.
4. The specific heat is assumed to be constant.

It is known that the last assumption is untrue even for air, and still more so for the mixed gases actually obtaining in the cylinder. In fact, the specific heat increases with rise of temperature which materially affects the result. Hence, for this and other reasons, the efficiency of a real engine will always fall short of the air standard, to which we shall return later.

It will be clear that for a given amount of heat supplied to the cylinder, the engine must convert *all* that

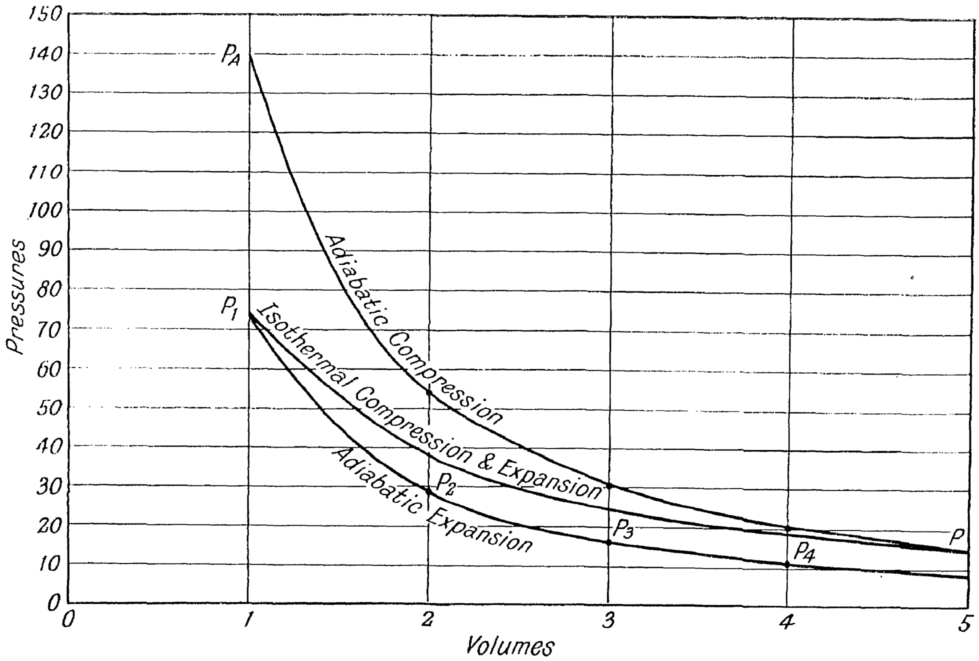

Pressures
150
140
130
120
110
100
90
80
70
60
50
40
30
20
10
0
0
1
2
3
4
5
Volumes
P_A
P_1
P_2
P_3
P_4
P
Adiabatic Compression
Isothermal Compression & Expansion
Adiabatic Expansion

FIG. 4

heat into mechanical work, if perfect efficiency of 100 per cent is to be attained; in other words, whatever the initial temperature of the gas, its final temperature would have to be absolute zero—a condition which is manifestly impossible. It is the difference between the initial and final temperature which determines the efficiency, but it will presently be shown that this difference is governed by the ratio of expansion (and compression), and if r = this ratio, we shall have the equation for air standard efficiency

$$E = 1 - \left(\frac{1}{r}\right)^{\gamma-1}$$

and since $\gamma - 1 = 0{\cdot}4$ we can write the equation

$$E = 1 - \left(\frac{1}{r}\right)^{\text{[illegible]}}$$

We are now able to calculate the compression and expansion curves of any engine.

Let V = the initial volume of a gas.

V_1 = the final volume.

P = the initial absolute pressure.

P_1 = the final absolute pressure.

T = the initial absolute temperature.

T_1 = the final absolute temperature.

Then $P_1 = P \times \left(\frac{V}{V_1}\right)^{\gamma}$

So if air at atmospheric pressure is rapidly compressed to one-fourth of its volume, this being a compression ratio of 4, it follows that

$$P_1 = 14{\cdot}7 \times 4^{1\cdot4}$$

now $4^{1\cdot4} = \log 4 \times 1{\cdot}4, = {\cdot}6020 \times 1{\cdot}4, = {\cdot}8428,$

$= \log 6{\cdot}963$

so $P_1 = 14{\cdot}7 \times 6{\cdot}963 = 102{\cdot}3$ lb. sq. in. abs.

The intermediate points can be found by the same method, and a curve plotted from these values is shown in Fig. 5.

The converse of this is true, and if air at absolute pressure of 102·3 lb. sq. in. be suddenly expanded in

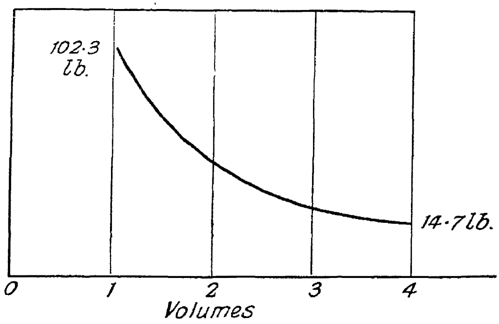


FIG. 5

a working cylinder to four times its volume, its final pressure will be 14·7 lb. sq. in.

To find the temperature after compression, we use the formula

$$T_1 = T \times \left(\frac{V}{V_1}\right)^{\gamma-1}$$

and at the same compression ratio of 4 to 1 this becomes

$$T_1 = T \times (4)^{0\cdot4}$$

we assume the initial temperature is 60° F., then on the absolute scale this becomes $60 + 460 = 520°$ F. abs.

so $$T_1 = 520 \times 4^{0\cdot4}$$

$4^{0\cdot4} = \log 4 \times 0\cdot4, = \cdot6062 \times 0\cdot4, = \cdot24080 = \log 1\cdot74$

then $$T_1 = 520 \times 1\cdot74 = 904\cdot8° \text{ F. abs.}$$

(444·8° F. on the ordinary scale.)

The converse of this is, of course, true, so that if air at a temperature of 904·8 abs. is suddenly expanded to four times its volume, its temperature will fall to 520 abs.

It will be a useful exercise for the student to calculate pressure curves for air compressed to all compression

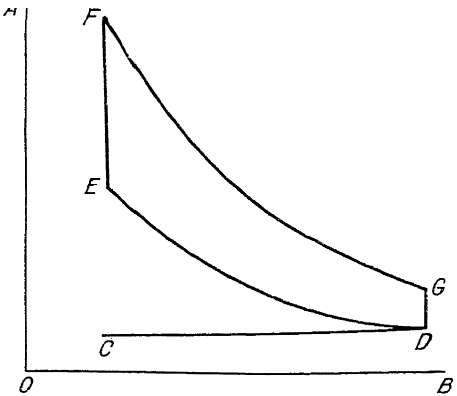


FIG. 6

ratios up to, say, 12, and to plot the results in an accurate graph which can be kept for reference.

A theoretical indicator diagram from a four-cycle petrol engine is shown in Fig. 6. In this *OA*, *OB* are lines of zero volume and zero pressure respectively; *CD* is the suction stroke when the cylinder is filled with air and fuel. *DE* is the compression curve, at *E* explosion is caused, and from *E* to *F* the pressure rises without increase in volume; *FG* is the expansion curve, and at *G* the opening of the exhaust allows the pressure to drop to that of the atmosphere, and *DC* represents this time the exhaust stroke.

Between *E* and *F* the gas mixture undergoes a chemical change which completely alters its nature,

and the expansion is performed with a mixture quite different from that which obtained during compression.

Generally a slight change of specific volume of the gas occurs after combustion, so that when brought to the same pressure and temperature, the burnt products occupy a different (sometimes less, sometimes greater) volume to what they did before ignition; and while in

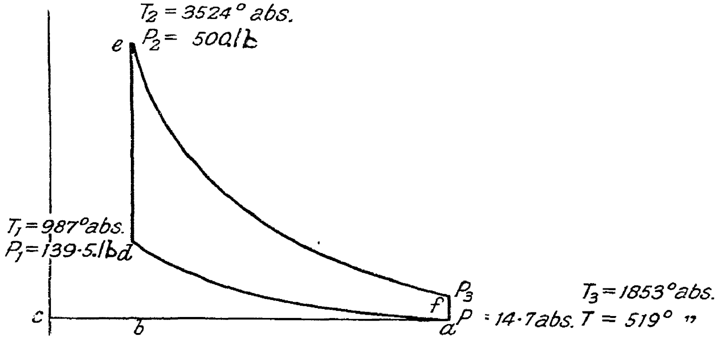


FIG. 7

a petrol engine this change is so small as to be negligible in ordinary calculations, it may be necessary to take it into account in special investigations. We will now proceed to calculate the pressures and temperatures which will occur in a petrol engine working with a compression of 5 to 1.

Fig. 7 shows the ideal diagram for an explosion engine of the four-cycle type, as in Fig. 6, both fuel and air being mixed in the cylinder before compression; it is assumed that ignition takes place at the end of the piston stroke, that is, on top dead centre. In the figure the horizontal line *ab* represents the piston stroke, and the line *bc* the clearance or compression space; hence *ac* represents the initial volume of the gas in the cylinder and *bc* its final volume; *bc* is one-fifth of *ac*.

Commencing with the piston at a, we assume the cylinder has been completely filled with the combustible mixture of petrol vapour and air at atmospheric pressure and temperature, viz., 14·7 lb. per sq. in. and 60° F., or $60 + 459 = 519°$ F. abs. As the piston moves inwards from a to b the gas is adiabatically compressed, and both the pressure and temperature rise until the point d is reached. Let V and V_1 represent the initial and final volumes respectively, and let $P\ P_1P_2P_3$ and $T\ T_1T_2T_3$ represent the pressures and temperatures at the four corners $adef$ of the diagram.

At the point a from which we start, we know the pressure is 14·7 lb. sq. in., and $T = 519°$ F. abs.

After compression the pressure $P_1 = P \times \left(\frac{V}{V_1}\right)^{\gamma}$ substituting our known values we have

$$P_1 = 14{\cdot}7 \times \left(\frac{5}{1}\right)^{1{\cdot}4}$$

$5^{1{\cdot}4} = \log 5 \times 1{\cdot}4, = 0{\cdot}698 \times 1{\cdot}4, = 0{\cdot}977, = \log 9{\cdot}484$

so $P_1 = 14{\cdot}7 \times 9{\cdot}484, = 139{\cdot}5$ lb. sq. in. abs.

or $139{\cdot}5 - 14{\cdot}7 = 124{\cdot}8$ lb. sq. in. above the atmosphere.

We now require to find the temperature after compression; this is given by the equation

$$\frac{T_1}{T} = \left(\frac{V}{V_1}\right)^{\gamma\text{-}1}$$

or $\quad T_1 = T \times \left(\frac{V}{V_1}\right)^{\gamma\text{-}1} = T \times 5^{0{\cdot}4}$

so $\quad T_1 = 519 \times 5^{0{\cdot}4}$ here $5^{0\ 4} = \log 5 \times 0{\cdot}4$

$\quad\quad = 0{\cdot}698 \times 0{\cdot}4 = 0{\cdot}279 = \log 1{\cdot}901$

then $T_1 = 519 \times 1{\cdot}901 = 987°$ F. abs. (or 987 459 $= 528°$ F. ordinary).

We find then that at d the gaseous mixture has a pressure of 139·5 lb. sq. in. abs., and a temperature of 987° F. abs. At this point ignition takes place, and the temperature, and consequently the pressure, are suddenly raised. The final temperature after combustion depends on the strength of the mixture, the heat value of the fuel, and its rate of burning when mixed with air. Up to the point where there is insufficient air (and therefore oxygen) to effect complete combustion, the richer the mixture the higher will be the temperature and pressure; for modern engines an explosion pressure of 500 lb. per sq. in. will be reached, but it is quite possible with a well-designed engine for the pressure to amount to nearly 600 lb. sq. in. The new temperature is found from the equation

$$\frac{T_2}{T_1} = \frac{P_2}{P_1} \text{ or } \frac{T_2}{987} = \frac{500}{140} \text{ or } T_2 = \frac{500 \times 987}{140} = 3524° \text{ F. abs.}$$

From this point the gases expand to the point f.

We have seen that at e $T_2 = T_3 \times \left(\frac{V}{V_1}\right)^{\gamma-1}$

or $$T_2 = T_3 \times 5^{0\cdot4} = T_3 \times 1\cdot901$$

and substituting the known value of $T_2 = 3524$ we have

$$3524 = T_3 \times 1\cdot901, \text{ whence } T_3 = \frac{3524}{1\cdot901} = 1853° \text{ abs.}$$

We also know that at e $P_2 = P_3 \times \left(\frac{V}{V_1}\right)^{\gamma}$

so $$P_2 = P_3 \times 5^{1\cdot4} = P_3 \times 9\cdot484$$

substituting the value of $P_2 = 500$

we have $$500 = P_3 \times 9\cdot484$$

whence $$P = \frac{500}{9\cdot484} = 52\cdot6 \text{ lb. sq. in. abs.}$$

At the point f the exhaust valve opens and the pressure drops to that of the atmosphere and the cycle is complete.

Any desired intermediate points on the compression and expansion curves may be calculated to enable the correct curves to be drawn, and we shall have obtained the ideal diagram for a petrol engine working under the conditions stated. Many factors combine to prevent the attainment of such ideal conditions, but, however nearly, by taking precautions, we may be able to approach an ideal diagram, it can easily be seen that the thermal efficiency of the engine is always disappointingly low. We have already remarked that to give an efficiency of 100 per cent the temperature at the end of expansion would have to be absolute zero, and we are now in a better position to appreciate the reason for this. Fig. 7, it will be remembered, is a pressure volume diagram, let us construct another, but this time making the ordinates or vertical scale to represent temperatures while the horizontal scale remains volumes as before. Fig. 8 is such a diagram, and it is at once seen from this how relatively high is the temperature at the point B where exhaust takes place—in other words, how much heat, unconverted into mechanical work, remains in the gases after expansion.

The actual efficiency of the engine is the ratio of the heat used to that of the heat supplied, or

$$\frac{\text{Heat supplied} - \text{heat rejected}}{\text{Heat supplied}}$$

or if we write H for the total quantity of heat supplied at the commencement of the cycle and H_1 for the quantity of heat rejected by the engine, then the efficiency E of the engine is given by the equation

$$E = \frac{H - H_1}{H}$$

If we refer to Fig. 8 we see that the heat is supplied from T_1 to T_2, that is, along the line DA while the volume remains constant, heat is also rejected from T_3 to T along the line BC where the volume is again

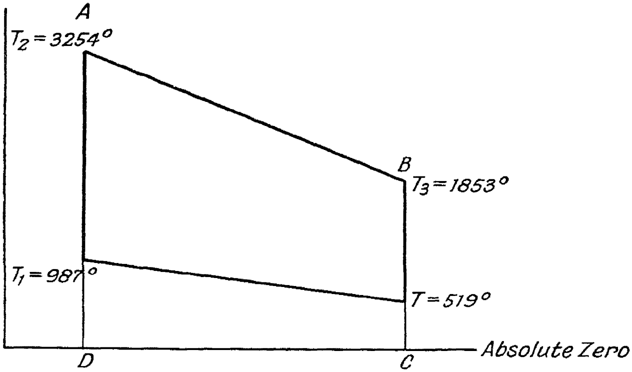


FIG. 8

constant, hence we can say that the *heat is both supplied and rejected at constant volume.*

Let us make K_v the specific heat of air at constant volume, then

the heat supplied is $H = K_v(T_2 - T_1)$
the heat rejected is $H = K_v(T_3 - T)$

so the efficiency E is therefore

$$E = \frac{K_v(T_2 - T_1) - K_v(T_3 - T)}{K_v(T_2 - T_1)}$$

or $E = 1 - \dfrac{T_3 - T}{T_2 - T_1}$ and substituting $E = 1 - \dfrac{1853 - 519}{3524 - 987}$

and $E = 1 - \cdot 525 = 0\cdot 475$, or $47\cdot 5$ per cent.*

* NOTE.—In practice, efficiencies of 30% are approached.

It follows that the efficiency of an explosion or constant volume engine is only dependent on the compression ratio, and is independent of the maximum and the initial temperature of the gases, and so long as the clearance volume is known the efficiency can be determined by the use of the simple formula

$$E = 1 - \left(\frac{V_1}{V}\right)^{\gamma-1}, \text{ or } E = 1 - \left(\frac{1}{r}\right)^{\gamma-1}$$

which we have already seen is the equation for the air standard efficiency.

Diesel Engines. So far we have confined our attention to the constant volume type of engine, to which class all petrol engines belong. We shall now turn our attention to the second or constant pressure type, otherwise known as the Diesel type. From the circumstance that such engines are self-igniting, electric ignition being dispensed with and the fuel being fired by the temperature attained during compression, they are also known by the somewhat clumsy name of Compression Ignition, or "C.I." engines. In these engines air alone, that is unmixed with fuel, is compressed in the cylinder, and the liquid fuel is admitted during the expansion stroke, the fuel being injected at such a rate that the pressure remains constant, the air being heated by the burning fuel during the first part of the stroke and so maintaining the pressure.

When the fuel is cut off, expansion commences, and continues throughout the remainder of the stroke. In a petrol or constant volume type of engine, the fuel and air are drawn into the cylinder together and compressed as one gas which forms a highly explosive mixture; we have seen, too, how, at the end of the compression stroke the temperature of the gas has reached a high value. In the case we examined the compression ratio was 5 to 1, and much higher temperatures

would have been reached had the compression been carried further, with the result that the explosive mixture would ignite spontaneously, producing the phenomenon of detonation, which, besides being destructive in its mechanical effects, makes at the same time for inefficiency. In a Diesel engine, on the other hand, air alone is compressed, and as there is no explosive mixture until such time as we chose to make one by injecting fuel, we can carry the compression, at the same time raising the temperature, as high as we have a mind, with the several important accruing advantages as will be seen later.

An important fact to remember is that while the pressure is kept constant during the admission of the fuel, it is far otherwise with the temperature which, during the process of combustion, rises considerably, with important effects on the efficiency and performance of the engine, and while the efficiency of the constant volume engine depends alone upon the compression ratio, the efficiency of the constant pressure engine depends partly on the compression ratio and partly on the temperature of the gas in the cylinder before expansion. It must, however, always be remembered that the hotter the gas before expansion, the greater will be the temperature at which exhaust commences, with an adverse effect on the efficiency of the engine.

In a Diesel engine, the point of the stroke at which the fuel is cut off can be altered to suit the conditions of load, and may vary through considerable limits, from, say, 5 per cent to 20 per cent of the stroke. We will assume a compression ratio of 12 to 1, which is quite normal in such an engine, and in order to study the effect of varying the period of fuel admission, we will take three examples in which the cut-off takes place at $\frac{2}{11}$ of the stroke, $\frac{1}{11}$ of the stroke, and $\frac{1}{22}$ of the stroke respectively. If we turn to Fig. 9 we see that

$\frac{1}{22}$ of the stroke is equal to $\frac{1}{24}$ of the total volume, and this added to $\frac{1}{12}$ of the total volume (the volume of the clearance space) gives—

$$\frac{1}{12} + \frac{1}{24} = \frac{2}{24} + \frac{1}{24} = \frac{3}{24} = \frac{1}{8} \text{ of the total.}$$

Starting then from the point a with pure air at $P = 14{\cdot}7$ lb. sq. in. abs., and at $T = 519°$ F. abs. As

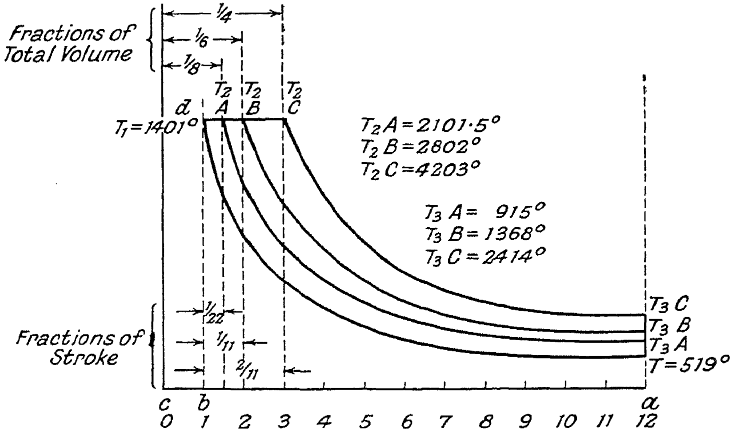


FIG. 9

the piston moves from a to b the air will be compressed, and at the point d the pressure P_1 will be found from the formula $P_1 = P \times r^{\gamma}$, or $P_1 = 14{\cdot}7 \times 12^{1{\cdot}4}$, proceeding as before—

log $12 = 1{\cdot}07918$, and $1{\cdot}07918 \times 1{\cdot}4 = 1{\cdot}5108$, $=$ log $32{\cdot}42$
so $P_1 = 14{\cdot}7 \times 32{\cdot}42$, $= 476$ lb. sq. in.

The temperature T_1 we get from the formula

$$T_1 = T \times r^{\gamma-1}, \text{ or } T_1 = 519 \times 12^{0{\cdot}4}$$

log $12 = 1{\cdot}07918$ and $1{\cdot}07918 \times 0{\cdot}4 = {\cdot}4316 =$ log $2{\cdot}70$
then $T_1 = 519 \times 2{\cdot}701 = 1401°$ F. abs.

Here the fuel is injected and combustion commences, the supply of fuel being maintained at such a rate as will keep the pressure constant until the point of cut-off is reached. During this process the temperature of the gas has risen considerably, and we require to know this. We saw from Charles' law that if the pressure remains constant the volume will vary directly as the temperature; we also see that during the period of combustion the volume has increased from 2 units to 3 units. The temperature T_2 which has caused this is found thus—

$$T_2 = T_1 \times 3/2 = 1401 \times \frac{3}{2} = 2101{\cdot}5^\circ \text{ F. abs.}$$

At this point expansion starts with an immediate lowering of the temperature, and the final temperature is found from the equation—

$$T_3 = \frac{T_2}{r^{\gamma-1}} = \frac{2101{\cdot}5}{8^{0{\cdot}4}} \text{ or } T_3 =$$

For our next example, we have the fuel cut-off at $\frac{1}{11}$ of the stroke, which we see from Fig. 9 is

$$\frac{}{12} + \frac{}{12} = \frac{}{6}$$

of the total volume. Up to the point d there is no change from the previous case, and here we have, as before, $P_1 = 476$ lb. sq. in. and $T_1 = 1401$, but in this case, during combustion the volume has increased from 1 unit to 2, hence

$$T_2 = T_1 \times \frac{2}{1} = 1402 \times 2 = 2802^\circ \text{ F. abs.}$$

After expansion the final temperature T_3 will be

$$T_3 = \frac{T_2}{6^{0{\cdot}4}} = \frac{2802}{2{\cdot}048} = 1368^\circ \text{ F. abs.}$$

Lastly, we have the fuel cut-off at $\frac{2}{11}$ of the stroke which Fig. 9 shows to be $\frac{3}{12}$ or $\frac{1}{4}$ of the total volume.

As before, $P_1 = 476$ lb. sq. in. and $T_1 = 1401°$ F. abs., while during combustion the volume has increased from 1 unit to 3.

So $T_2 = 1401 \times 3 = 4203°$ F. abs., and the final temperature, after expansion, is

$$T_3 = \frac{T_2}{4^{0\cdot4}} = \frac{4203}{1\cdot741} = 2414° \text{ F. abs.}$$

Summarizing these results we have obtained

Cut-off Stroke	Cut-off Total Volume	T	T_1	T_2	T_3	$E\%$[1]
A . . $\frac{1}{22}$	$\frac{1}{8}$	519	1,401	2,101	915	59·3
B . . $\frac{1}{11}$	$\frac{1}{6}$	519	1,401	2,802	1,368	56·7
C . . $\frac{2}{11}$	$\frac{1}{4}$	519	1,401	4,203	2,414	51·7

It remains for us to calculate the efficiency under the three conditions; the process is somewhat different from that in the case of the constant volume cycle, because the heat, as we have seen, is taken in at constant pressure, and discharged at constant volume, instead of being both taken in and discharged at constant volume. Here we recall the difference between the specific heats of air at constant pressure and constant volume. Let H = the heat added at constant pressure, and H_1 = the heat discharged at constant volume.

Then $\quad H = K_p(T_2 - T_1)$ and $H_1 = K_v(T_3 - T)$

and the efficiency

$$E = 1 - \frac{K_v(T_3 - T)}{K_p(T_2 - T_1)}$$

here $\quad \frac{K_v}{K_p} \quad \frac{1}{\gamma}$

[1] Values of E obtained from subsequent calculation.

so the equation simplifies into

$$E = 1 - \frac{(T_3 - T)}{\gamma(T_2 - T_1)}$$

and substituting the values we have just found we have in case (A) $\frac{1}{8}$ cut-off

$$E = 1 - \frac{(915 - 519)}{1{\cdot}4\ (2101{\cdot}5 - 1401)}$$

$$= 1 - \frac{396}{1{\cdot}4\ (700{\cdot}5)}$$

$$E = 1 - \frac{396}{980{\cdot}7} = 1 - 0{\cdot}403 = 0{\cdot}593 \text{ or } 59{\cdot}3 \text{ per cent.}$$

In case (B) $\frac{1}{6}$ cut-off

$$E = 1 - \frac{(1368 - 519)}{1{\cdot}4\ (2802) - 1401)}$$

$$= 1 - \frac{849}{1{\cdot}4 \times 1401}$$

$$= 1 - 0{\cdot}433 = 0{\cdot}567 = 56{\cdot}7 \text{ per cent.}$$

In case (C) $\frac{1}{4}$ cut-off

$$E = 1 - \frac{(2414 - 519)}{1{\cdot}4(4203 - 1401)}$$

$$= 1 - \frac{1895}{1{\cdot}4 \times 2802}$$

$$= 1 - 0{\cdot}483 = 0{\cdot}517 = 51{\cdot}7 \text{ per cent.}$$

These percentages have been added as the last column of the table, and Fig. 10 has been plotted to show this variation graphically.

It is instructive to compare these efficiencies with an engine operating on the constant volume cycle with the same compression ratio. In that case

$$E = 1 - \left(\frac{1}{r}\right)^{\gamma - 1} \text{ or } E = 1 - \left(\frac{1}{12}\right)^{0{\cdot}4}$$

which we find to be 63 per cent, but in this case the maximum pressure would be $476 \times 4203/1401 = 1428$ lb. sq. in. We see from this that *for equal ratios of compression*, the constant pressure or Diesel type has no advantage over the constant volume (petrol) type of engine.

In the cases we have hitherto examined, we have

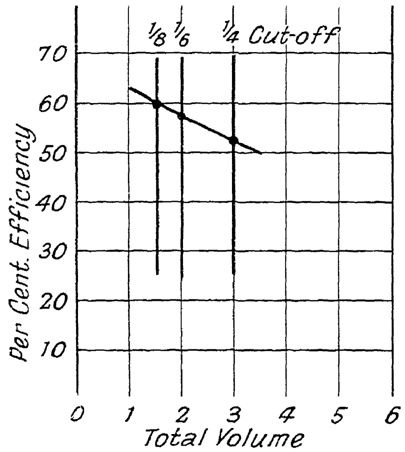


FIG. 10

considered the Diesel engine as rejecting its residual heat at constant volume, as in practice it always does, but it is instructive to imagine another case where the rejection of heat takes place at constant pressure; for this we must further suppose that the working substance is expanded adiabatically down to the pressure of the atmosphere before it is discharged into the exhaust. These conditions cannot be realized in practice, but were this possible, we should have a further type of engine both receiving and rejecting its heat at constant pressure. Fig. 12 is an ideal diagram from

such an engine in which compression commences at A and continues to B, where fuel is admitted and combustion takes place during the volumetric change from

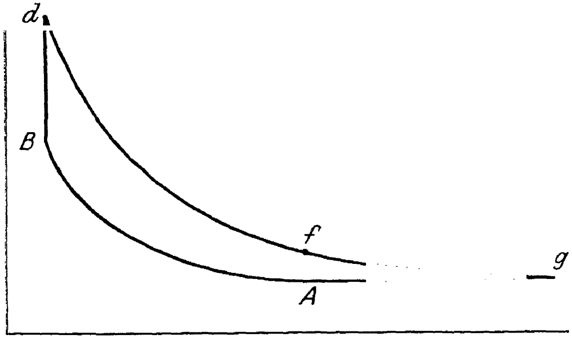


FIG. 11

B to C. C to D is the adiabatic expansion curve, which is prolonged until D falls to atmospheric pressure and exhaust takes place from D to A. The impracticability

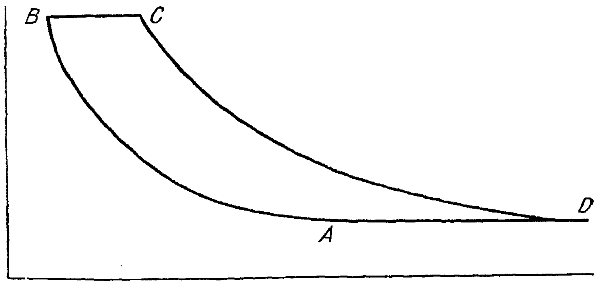


FIG. 12

of the cycle consists in the fact that the expansion stroke is much longer than the compression stroke, so that compression must be carried out by some other

means than in the working cylinder. An engine on these lines might be arranged to work either the constant volume or the constant pressure cycle, and it is useful to see the effect on the efficiency.

We will first take the example we have already worked for the constant volume type (Fig. 7); here we found the following conditions—

Compression ratio 5 to 1.

At commencement of compression—

	$T = 519°$ abs.	$P = 14{\cdot}7$
After compression	$T_1 = 987°$ abs.	$P_1 = 139{\cdot}5$
After explosion	$T_2 = 3524°$ abs.	$P_2 = 500{\cdot}0$
After expansion	$T_3 = 1853°$ abs.	$P_3 = 52{\cdot}6$

But supposing that if instead of releasing the gases to the exhaust at the point f we continue the expansion until the pressure had dropped to 14·7 lb. abs. at the point g (Fig. 11) so that the expansion line met that of the atmosphere, we can find the number of expansions from the equation

$$r = \left(\frac{500}{14{\cdot}7}\right)^{\frac{1}{\gamma}} = 34^{0{\cdot}71} = 12{\cdot}3 \text{ expansions.}$$

We next find the temperature at the point g, which we get from the equation

$$T_3 = T_2/12{\cdot}3^{\gamma-1} = \frac{3524}{2{\cdot}728}$$

whence $\quad T_3 = 1290°$ F. abs.

We can now calculate the efficiency, since we know that the heat supplied to the cycle at constant volume is

$$H = K_v(T_2 - T_1)$$

and the heat discharged is $H_1 = K_p(T_3 - T)$
it follows that

$$E = \frac{K_v(T_2 - T_1) - K_p(T_3 - T)}{K_v(T_2 - T_1)}$$

from this

$$E = 1 - \gamma\left(\frac{T_3 - T}{T_2 - T_1}\right)$$

$$= 1 - 1{\cdot}4\left(\frac{1290 - 519}{3524 - 987}\right)$$

$$= 1 - 1{\cdot}4 \times \frac{771}{2537} = 1 - 0{\cdot}425$$

So $\quad E = {\cdot}57$ or 57 per cent.

We now take the case of a Diesel engine (Fig. 12) where the diagram is based on the second example worked out, when the fuel is admitted for one-sixth of the total volume, but in this case we assume the expansion carried down to atmospheric pressure, and reference to Fig. 9 shows that the gases will have to be expanded $6 \times 2 = 12$ times their volume. The temperature then will be

$$T_3 = T_2/12^{\gamma-1}, = 2802/2{\cdot}706, = 1031^\circ \text{ F. abs.}$$

and as in a former case both compression and expansion are between the same limits of pressure, and are adiabatic; hence the efficiency will depend only on the degree of compression, and is found from the equation

$$E = 1 - \left(\frac{1}{r}\right)^{\gamma-1}$$

and as $r = 12$

$$E = 1 - \left(\frac{1}{12}\right)^{0{\cdot}4}$$

or $\quad E = 1 - 0{\cdot}37 = 63$ per cent,

so that by expanding down to atmospheric pressure the efficiency has been increased from 56·7 to 63 per cent, and we note that in this case the efficiency is only dependent upon the compression ratio, and provided some means are devised to expand the gases down to the atmospheric line, the effect of the length of the burning period, which is so clearly marked in the Diesel cycle, disappears, and we have the interesting fact that the Air Standard formula applies to—

(*a*) The constant volume type of engine, where heat is both received and rejected at constant volume.

(*b*) The constant pressure type, where heat is both received and rejected at constant pressure, but it does not apply to the Diesel type where heat is received at constant pressure and rejected at constant volume.

Fig. 13 gives a temperature volume diagram for a Diesel engine, in which the three conditions of fuel admission, illustrated in Fig. 9, are followed (conditions A, B, and C of the table on page 30). In this, as in Fig. 8, the vertical lines represent absolute temperatures, and indicate clearly the large amounts of heat added during the injection of the fuel, and how relatively small a percentage of this can be converted into work before the end of the stroke is reached, which accounts for the drop in efficiency of the order of 20 per cent in the case of C as compared with A, as shown under the heading of column "E" in the table.

It is evident from this that the very high initial temperatures before expansion are to be avoided as far as possible, alike on the score of efficient performance, and of the destructive effect of such temperatures on the working parts of the mechanism.

To summarize the foregoing—

1. In the constant volume type engine, if the gases are expanded to the same volume as before

compression, the air standard efficiency is dependent on the compression ratio only.

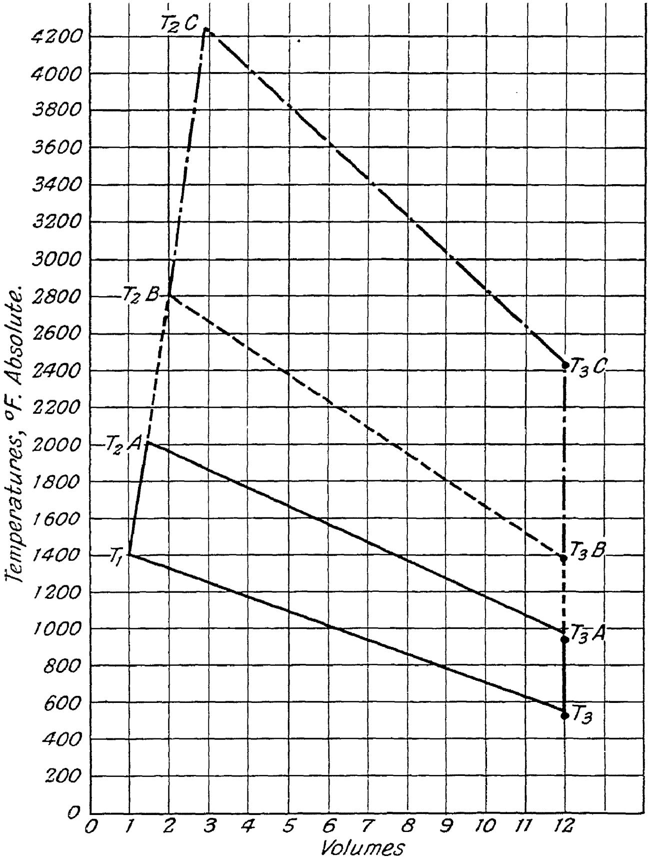


FIG. 13

2. In the constant volume type, if the gases are expanded to atmospheric pressure, the efficiency is

dependent partly on the compression ratio, and partly on the maximum temperature, and with an increase of temperature it increases slightly.

3. In the constant pressure type, if the gases are expanded to the same volume as before compression, the efficiency is mainly dependent on the compression ratio, and to some extent on the maximum temperature, and with increase of temperature it decreases.

4. In the constant pressure type, if the gases are expanded to atmospheric pressure, the efficiency is only dependent on the compression ratio, being entirely independent of maximum temperature and pressure.

5. For equal compression ratios, the constant pressure type expanded to atmospheric pressure, and the constant volume type expanded to original volume, have precisely the same efficiency.

It must be again emphasized that these results are based on the following conditions—

1. The working substance is pure, dry air.

2. The specific heat remains constant at constant volume irrespective of the temperature.

3. Both compression and expansion are truly adiabatic.

4. That in the constant volume type combustion is absolutely instantaneous.

5. That in the constant pressure type combustion is so regulated as to maintain exactly constant pressure.

6. That the gases are expanded to the same volume as they originally occupied.

In practice none of these conditions are fulfilled, except perhaps approximately the last.

For convenience we may tabulate the five conclusions as in the tables opposite.

These have been drawn as a graph in Fig. 14, where it is seen that while at first the efficiency increases very rapidly with the compression ratio until the ratio of

Type	Expansion to	Air Efficiency Depends on
Constant Volume	*Original Volume*	*Compression Ratio*
Constant Volume	Atmospheric Pressure	Compression ratio and maximum temperature increasing with temperature
Constant Pressure	Original Volume	Compression ratio and maximum temperature decreasing with temperature
Constant Pressure	Atmospheric Pressure	Compression ratio
Constant Volume Constant Pressure	Original Volume Atmospheric Pressure	Same efficiency if same compression ratio

TABLE OF AIR STANDARD EFFICIENCIES

Compression Ratio	Air Standard Efficiency
2	0·242
3	0·356
4	0·426
5	0·475
6	0·511
7	0·541
8	0·565
10	0·602
15	0·661
20	0·698

about 5 to 1 is attained, thereafter there is a marked falling off in the rate of efficiency increase, so that at about 12 to 1 the advantage of increasing the compression becomes relatively slight.

From what has already been said, it is evident that the

ability to convert heat into work depends on the lowering of the temperature from a hot source to a colder receiver, hence it is only through the existence of a difference in temperature that the conversion of heat into work becomes possible, and if all substances were at the same level of temperature, no matter how high, no mechanical effect could be produced. It is, furthermore, impossible to convert the whole of any heat supply into work, since, as we have seen, the initial heat must be measured on the absolute temperature scale, and to make use of it all implies the existence of a body at the absolute zero of temperature as the receiver into which the heat could be rejected. So it comes about that if heat is taken in at T_1 and rejected at T, the maximum efficiency can only be obtained when *all* the heat is supplied at T_1 and *all* is rejected at T, and with such limiting temperatures it is essential that no heat is received below T_1 and none rejected above T, and in planning the design of any engine it is

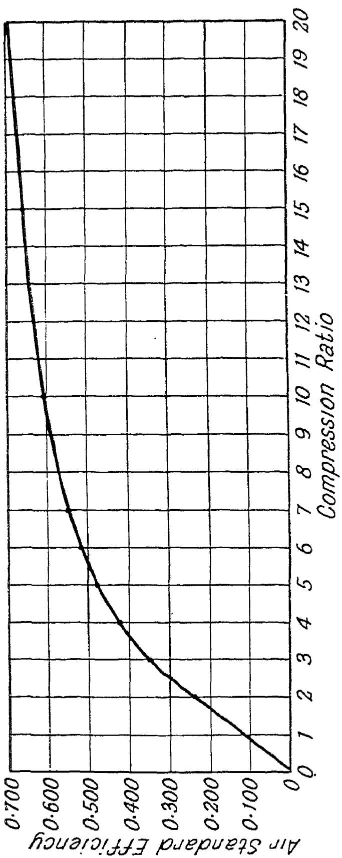


FIG. 14

of the first importance to be sure that these conditions are not lost sight of.

Finally, the conditions of practical working modify to a very material extent the results as compared with those arrived at from theoretical considerations, for the following reasons—

1. The working substance is not air, but a mixture of gases whose specific heats at constant pressure and constant volume are not in the same ratio as those of air.

2. It is now known that the specific heat of the mixed gases is not constant, but increases materially at high temperatures.

3. The researches of the Gaseous Explosions Committee of the British Association have shown that the internal energy of the normal mixture of gases is considerably higher than that of air, amounting from about 25 per cent at 1,500° F. to about 50 per cent at 3,500° F. But this depends upon the proportions of the gases present in the cylinder, which may be normally

	%
N and O_2	83
CO_2	5
H_2O	2
	100

4. Neither compression nor expansion can be strictly adiabatic, as there must exist some small interchange of heat between the gases and the cylinder, which, however, decreases with increase in the speed of the engine. The variation in specific heat also has a bearing on this.

5. In an explosion engine (constant volume type) the combustion must occupy *some* period of time, and is therefore never instantaneous in the strict sense.

6. In practice, the exhaust valve must always open

before the end of the expansion stroke, hence the expansion is not taken down to exactly the original volume.

Ricardo has pointed out that although the air standard efficiency of explosion engines is independent of the maximum temperature, it will be found that other things being equal, the lower the maximum temperature the higher will be the efficiency, when allowance is made for the alteration in the specific heat.

SECTION II

FUEL TECHNOLOGY

BY

H. KERR THOMAS, M.I.MECH.E., M.I.A.E.

SECTION II

FUEL TECHNOLOGY

Varieties of Fuel. In the foregoing section we have dealt at some length with the theoretical aspect of the conversion of the fuel heat into work, in an internal combustion engine, and we must never lose sight of the main principles which have emerged from the discussion. We have seen that from 20 to 30 per cent of the heat supplied to the engine can be converted into useful work in a normal well-designed engine, and we will now devote some space to the consideration of the fuel itself, and how its special characteristics must be studied if we are to take advantage of its favourable qualities, and at the same time steer clear of difficulties arising from its defects.

For practical purposes we may divide the liquid fuels into two classes, the volatile and non-volatile hydrocarbons; the former being used in engines of the constant volume type, with electric ignition, and the latter in constant pressure or Diesel type engines, in which the ignition is brought about by the very high temperatures induced by compression.

We need not go into the details of heavy fuels for use in the latter type of engine, as Diesel engines can be operated on a wide range of heavy fuels, in fact, from a comparatively light mineral oil to tar, and to deal with these would encroach too far on the space at our disposal. Moreover, a great deal of experimental work needs to be carried out before anything like the same information is available, as in the case of the volatile fuels: in the ensuing pages, therefore, only the latter will be considered.

At the outset it may be stated that the power and efficiency of a petrol engine may appear to vary through wide limits with the type of fuel used, but as will be seen as we proceed, in a *suitable* engine this observation does not apply, since it is possible to design, if necessary, to suit the special characteristics of almost any volatile fuel, provided it is one which on burning produces complete combustion, and does not form excessive tarry products which will deposit as carbon on the interior surfaces of the engine. We know from thermodynamic considerations that the only factor affecting the thermal efficiency is that of the compression ratio to be employed, and for general purposes a compression ratio of from 5·0 to 6·0 to 1 is found; for this, there is a wide range of suitable fuels on the market and commercially available. If for special purposes very high compressions are needed, it is possible to compound certain substances with the fuel which will enable them to be used without detonation or pre-ignition, which are the two limiting factors to any ratio of compression. Certain fuels of high volatility may be added to facilitate the starting of engines in cold weather, and this is at the present time a common variation practised by the large suppliers.

The common supposition that some engines will give greatly increased power with certain fuels cannot be substantiated by experimental evidence, and where any indication of this is alleged, it is due either to the fuel working near the detonation point, necessitating running with an unduly retarded spark, or else, that in the engine in question, some characteristic of its design tends to produce imperfect distribution among its cylinders, in which case, certain fuels may be prone to condensation and deposition of liquid in parts of the induction system, and this factor will not be considered in this section.

Volatile Fuels. The volatile fuels commercially available at the present day consist of petrol, benzol, kerosene, and alcohol. Petrol normally is distilled from crude petroleum and consists of all the fractions which boil between the temperature limits of 140° F. and, say, 400° F.

There are, chemically, three series to which these fractions belong, viz.—

The paraffins	C_nH_{2n+2}
The naphthenes	C_nH_{2n}
The aromatics	C_nH_{2n-6}

In addition to these main groups there are commonly a small proportion of what are known as the olefine series, and this proportion varies according to the process by which the petrol is obtained. What may be termed "natural" petrol is obtained by simple distillation from the crude oil, while what is known as "cracked" petrol, is produced by the so called cracking process, where the crude oil, under great pressure, is subjected to very high temperatures, by which actual decomposition of the crude base occurs: with the increased demand for petrol the latter process has latterly become much more general.

In view of the important effect the fuel has upon the combustion of the gases, it will be necessary to study its composition in some detail. The main series classified above may be further subdivided as shown in Table I which follows. In this have been included the boiling points and the specific gravities of each of the substances named.

It is seen that there is a wide difference in the composition of various petrols, and it should be specially noted that the specific gravity of the fuel gives no indication of its composition; moreover, as a guide to the relative value of the fuel, the specific gravity is

worse than useless, and the commonly heard statement that fuels of relatively high specific gravity are necessarily inferior is quite erroneous. Conversely, a low specific gravity by no means indicates advantages. Hexane, for example, in the paraffin series, with its specific gravity of ·663 would, for reasons we shall soon encounter, be an entirely unsuitable fuel for general use.

TABLE I

COMPOSITION BOILING POINT IN DEGREES F. AND SPECIFIC GRAVITY AT 60° F. VARIOUS FUELS

FUEL	Symbol	Boiling Point	Specific Gravity
Paraffin Series—		° F.	
Hexane	C_6H_{14}	156	·663
Heptane	C_7H_{16}	209	·691
Octane	C_8H_{18}	258	·709
Nonane	C_9H_{20}	302	·723
Decane	$C_{10}H_{22}$	343	·735
Undecane	$C_{11}H_{24}$	383	·746
Naphthene Series—			
Cyclohexane	C_6H_{12}	178	·780
Hexahydrotoluene	C_7H_{14}	212	·770
Hexahydroxylene	C_8H_{16}	246·2	·756
Aromatic Series—			
Benzene	C_6H_6	176	·884
Toluene	C_7H_8	230	·870
Xylene	C_8H_{10}	284	·862

As a rough rule (with some exceptions, however), the paraffin series predominate in commercial petrols, but this is largely influenced by the locality from which the crude oil is derived as the following analysis of seven commercial petrols exhibits. (See Table II.)

If the petrol consisted solely of the paraffin series, the specific gravity would be some measure of its volatility, there are, however, always present a proportion,

generally small, of the aromatics, and the high specific gravity of these, as shown in Table I, completely upsets the conclusions which might be drawn from the specific gravity of the mixed fuel alone. Of the fuels analysed in Table II, the fuels giving the best results as to power and efficiency are Nos. 1, 4, and 7, while No. 5 is the least satisfactory in these respects.

TABLE II

ANALYSIS OF TYPICAL COMMERCIAL PETROLS

	1	2	3	4	5	6	7	Average
Paraffins	26	62	61	38	68	80	10	49·3
Naphthenes	35	23	30·5	47	20	15·2	85	36·5
Aromatics	39	15	8·5	15	12	4·8	5	14·2
Total	100	100	100	100	100	100	100	100
Specific Gravity	·782	·723	·727	·760	·719	·704	·767	·740

The exhaustive tests carried out by Ricardo show that of the three categories the aromatics are the most desirable constituent, the naphthenes the next, while the paraffins are the most undesirable, and the smaller the proportion of them, the better the results.

What is known as detonation is the setting up within the cylinder of an explosion, due to causes to be examined later, and the phenomenon of detonation is the most valuable factor in estimating the quality of a fuel, for as we saw in Section I, a high ratio of compression is in every way advantageous, this ratio becomes impossible if the fuel detonates—or even tends to detonate at the pressure desired. Detonation primarily depends on the chemical composition of the fuel, and the paraffin series are all the worst from this aspect, and become worse still as their specific gravity

increases—hexane being better than heptane, etc.: the naphthenes are much better in this respect, and the aromatics are the best of all.

Commercial benzol is a spirit distilled from coal tar, and consists mainly of pure benzene, C_6H_6, with a certain proportion of toluene and traces of xylene, all of which are aromatics; its specific gravity varies from ·875 to ·882, depending on the amount of toluene it contains. Benzol is a better fuel than petrol, but, to obtain the full advantages of its use, a higher compression ratio is necessary. In practice benzol is generally mixed with petrol in something like equal proportions.

The fuels belonging to the group of alcohols are methyl-, ethyl-, and butyl-alcohols; they cannot correctly be classed with the hydrocarbons as they all contain oxygen. From the point of view of detonation, they are actually better than the aromatics but under high compression methyl-alcohol is liable to pre-ignite. They have a high latent heat and a low flame temperature, with the result that the engine operates at a high thermal efficiency; the maximum output of power is greater, and the flow of heat to the cylinder walls is lower than is the case with either petrol or benzol.

In determining the value of a volatile fuel for use in an internal combustion engine, it is necessary to take into account the following characteristics—

1. Tendency to detonate.
2. Latent heat
3. Volatility.
4. Calorific value of the fuel itself.
5. Heat value of the air fuel mixture.

It is an important fact that when vaporized and mixed with the required amount of air for complete combustion, all volatile liquid fuels have practically the same heat value *per cubic inch of the mixture* drawn into the cylinder, and consequently, under the same

conditions, they will all produce the same power, and will show the same thermal efficiency. Variation in power output will only be obtained by varying the compression, or altering the degree of vaporization in the carburettor.

Detonation. As previously noticed, the amount to which compression can be raised is determined by the conditions which cause detonation or pre-ignition. In the case of all petroleum fuels detonation takes place before pre-ignition, and therefore produces the latter, but in certain other fuels, e.g. ether, alcohol, and pure aromatics, pre-ignition may occur without preliminary detonation. It has already been stated that as regards detonation the paraffins are the worst and the aromatics the best. Mixtures of the two give detonation results in almost direct proportion to the relative quantities of the two in the mixture in question; so we find that the highest useful compression which can be employed is approximately as given in the following table with mixtures of heptane and benzol—

TABLE III

Mixture		Highest Useful Compression Ratio
Heptane	Benzol	
100	0	3·75–1
80	20	4·30–1
60	40	4·90–1
40	60	5·60–1
20	80	6·35–1
0	100	7·25–1

These values must not be taken as absolute, since there are other circumstances to be taken into account, notably the design of the combustion head, but the figures are sufficiently indicative of the beneficial effect of the aromatics.

As a measure of the behaviour of a fuel in regard to detonation, it is usual to describe it by what is known as its "toluene value." Standard petrol free from aromatics is taken as zero, and pure toluene as 100 at the other end of the scale; then in any sample under examination, the amount of toluene necessary to be added to "standard aromatic free petrol" to produce the same tendency to detonate as the sample possesses, furnishes a comparative measure known as the toluene value.

Thus a petrol might possess a highest useful compression ratio of 5·35 with a toluene value of 16·5 (highest useful compression ratio is generally abbreviated as h.u.c.)—which means that 16·5 per cent of toluene would have to be added to standard aromatic free petrol to delay detonation until at the same compression ratio as in the example.

Highest useful compression ratio is the highest ratio at which an efficiently designed engine can be operated without detonation at any mixture strength, and with any ignition timing. The researches of Ricardo have established the highest useful compression ratio and the toluene values for a large number of fuels, some of which values are given in the table on page 54.

To show the relationship between the toluene value and the h.u.c. the following diagram is plotted from the figures in the table (see Fig. 1). It may be stated that most engines will run at their maximum efficiency on a fuel which has a toluene value of from 15 to 20 per cent, and it is usual for commercial petrols to be maintained at an approximately constant toluene value.

Petrol is a spirit which burns slowly at atmospheric pressure, as a simple experiment will demonstrate, and the fact that in an engine ignition and combustion take place—for practical purposes—instantaneously, is due to the swirling or turbulence within the combustion

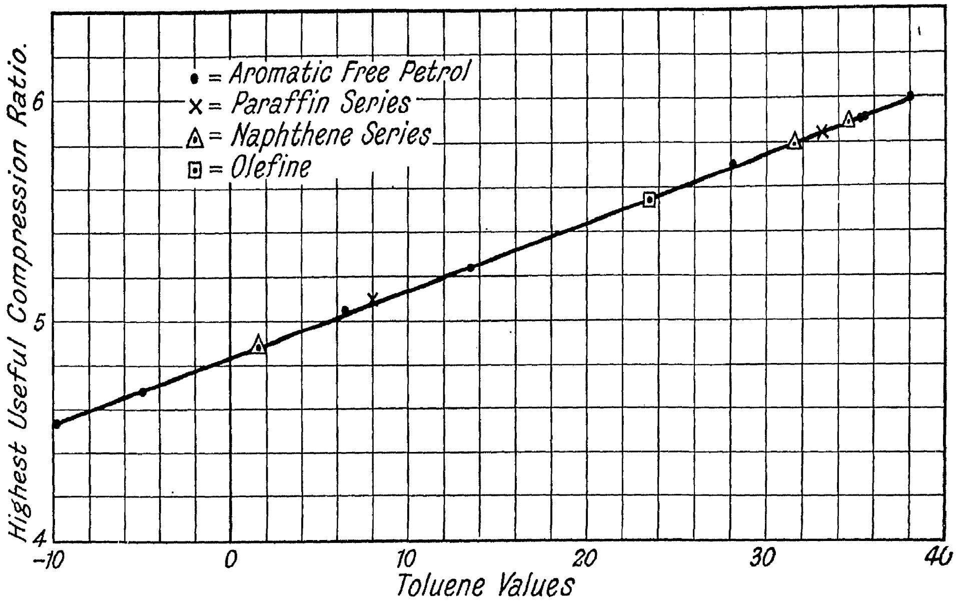

Highest Useful Compression Ratio.
4
5
6
-10
0
10
20
30
40
Toluene Values
• = Aromatic Free Petrol
X = Paraffin Series
△ = Naphthene Series
⊡ = Olefine

FIG. 1

TABLE IV (*Ricardo*)

FUEL	Highest Useful Compression Ratio	Toluene Value Toluene = 100%
Petrols—		
Aromatic Free Petrol .	4·85	0
Petrol "A" . . .	6·0	38
,, "B" . . .	5·7	28
,, "C" . . .	5·25	13·5
,, "D" . . .	5·35	16·5
,, "E" . . .	4·7	− 5·0
,, "F" . . .	5·05	6·5
Heavy Fuels—		
Heavy Aromatics . .	6·5	55
Kerosene . . .	4·2	−22
Paraffins—		
Pentane . . .	5·85	33
Hexane (80% pure) .	5·1	8
Heptane (97% pure) .	3·75	−37
Naphthenes—		
Cyclohexane . . .	5·90	35
Hexahydrotoluene . .	5·80	31·5
Hexahydroxylene . .	4·9	1·5
Olefine—		
Cracked Spirit . .	5·55	23·5
Aromatics—		
Benzene . . .	6·9	67
Toluene (99% pure). .	above 7·0	100
Xylene (91% pure) . .	above 7·0	85
Alcohols—		
Ethyl Alcohol (98%) .	above 7·5	above 88
Butyl Alcohol . .	7·3	80

space, caused by the rapid compression, and the form of the combustion head has much to do with the matter, as will be seen later when we come to discuss cylinder design. The circumstance that slow-burning fuel is less prone to detonate makes it more desirable.

Latent Heat. The next characteristic we have to consider in a fuel is its latent heat, as it has an important influence on the density of the charge drawn into the cylinder. We know from the Law of Charles that the volume of a given weight of gas varies with its absolute temperature. It follows that the *weight* of a given volume must vary *inversely* as its absolute temperature.

The latent heat of the fuel affects the result in two ways; first it determines its rate of evaporation, the lower the latent heat, the more rapid is the evaporation. Secondly, it has a more important effect; in any engine there is always a certain amount of pre-heating of the incoming gas mixture; in fact, this is specially provided for by arranging a "hot spot" in the intake manifold, where at some point or points, contact is made between the intake manifold and the exhaust pipe. The effect of this is to furnish sufficient heat to cause evaporation of the fuel, but as this heat becomes latent, there is not necessarily much, if any, perceptible rise in the temperature of the gases. This is very important, because to obtain the maximum weight of the charge of gas, we must, as we have just seen, keep its temperature as *low* as possible. As a result of the pre-heating, complete evaporation of the fuel almost always occurs before the gases enter the cylinder on the suction stroke, any unvaporized drops of liquid are instantly evaporated by contact with the hot cylinder or the hot residual exhaust gases.

The initial absolute temperature within the cylinder depends, therefore, upon the latent heat of evaporation of the fuel and the amount of heat applied during the passage of the gases through the intake manifold.

Ricardo has demonstrated that all normal fuels are completely evaporated before compression begins, and this has usually been caused by heating outside the cylinder; the heat so applied tends to be absorbed by the latent heat of evaporation, hence a relatively high latent heat is advantageous as tending to lower the absolute temperature of the gas and consequently to increase the weight of fuel taken in. The conclusion from this is that for a given amount of pre-heating, the volumetric efficiency (and consequently the power produced) will increase with the latent heat of the fuel.

We can, therefore, summarize thus—

(*a*) The power output is inversely proportional to the absolute temperature of the gases at the commencement of compression, since the temperature controls the weight of the charge, and consequently the volumetric efficiency.

(*b*) In similar circumstances the absolute temperature before compression depends on (1) the amount of external heating and (2) on the latent heat of the fuel; the volatility has little to do with the matter.

It is a fortunate circumstance that when the heat value of the fuel is relatively low, its latent heat is generally relatively higher, so that the actual amount of power developed does not materially vary as between different fuels.

Benzene is a good example of this; its total heat energy is appreciably lower than that of petrol (47·5 ft.-lb. per cubic inch as against 48·6 ft.-lb. for heptane), but the latent heat of benzene is much higher than that of petrol (172 B.Th.U. per lb. as against 133 B.Th.U. for heptane), consequently the power output for benzene is practically the same as that from petrol (within 1 per cent).

The case is even more marked in the example of alcohol. The latent heat of methyl-alcohol is 512 B.Th.U.

per lb., but the quantity of air required for combustion is much lower than is the case with petrol (6·44 to 1 as against 15 to 1), and in spite of the fact that the total heat energy is lower than that of either petrol or benzene, we actually obtain an increase in power as compared with these fuels.

Ricardo found that the volumetric efficiency of an engine using petrol and ethyl-alcohol, with correct air mixtures under similar conditions as to temperature and a compression ratio of 5 to 1 was 76 per cent in the case of petrol and 81 per cent in the case of alcohol, this being mainly due to the high latent heat of the latter. The considerable variation in the latent heat of various fuels is shown in the following table—

TABLE V

Fuel	Latent Heat of Evaporation (B.Th.U. per lb.)
Paraffin Series—	
Hexane	156
Heptane	133
Octane	128
Nonane	—
Decane	108
Naphthene Series—	
Cyclohexane	156
Hexahydrotoluene	138
Hexahydroxylene	133
Aromatic Series—	
Benzene	172
Toluene	151
Xylene	145
Olefine Series—	
Heptylene	167
Alcohol Group—	
Ethyl Alcohol	397
Methyl Alcohol	512

In the case of engines using kerosene as fuel, a considerable falling off in power output occurs as compared with petrol, since, in order to ensure evaporation, excessive heat must be applied in the induction system, resulting in an appreciable drop in volumetric efficiency which is not compensated for by the latent heat of evaporation. The result is a drop of some 15 per cent in power as compared with petrol.

Volatility. The volatility of any fuel naturally determines the amount of pre-heating required, and the amount of pre-heating in its turn determines the use we can make of the latent heat of the fuel, and it is of interest to note that the number of cylinders the engine has, has a bearing on the matter, since the more cylinders, the greater must be the length of the induction pipe, and to ensure even distribution and to prevent condensation in this, the more may pre-heating be necessary, and the rise or fall of temperature in the induction pipe gives some guide as to the volatility of the fuel, and experimental observation shows there is a wide difference between fuels in this respect.

Mixture Ratio. The ratio of air to fuel to bring about complete combustion varies within considerable limits for different fuels, and this has a corresponding effect on the temperature of the gases at the end of the suction stroke. As Table V shows, there is a wide variation in the latent heat of the various fuels, and if we take for illustration two examples, hexane with a latent heat of 156 B.Th.U. per pound, and toluene with a latent heat of 151 B.Th.U. per pound; 1 lb. of hexane will require 15·2 lb. of air for its combustion, while toluene, with nearly the same latent heat, only requires 13·4 lb. of air; hence in the case of toluene, the lesser weight of air will be cooled to a greater extent than the greater weight of air in the case of hexane. Thus "Toluene mixture" will be cooled through 40·5° F.,

while a "hexane mixture" will, under the same conditions, only be cooled through 37·8° F., and since, as we have seen, the weight of fuel increases inversely with the temperature of the gas, the volumetric efficiency obtainable with the toluene will always be greater, in the same circumstances than with hexane.

The following table gives the air to fuel ratio required for various fuels—

TABLE VI

Fuel	Air to Fuel Ratio by Weight
Paraffin Series—	
Hexane	15·2
Heptane	15·1
Octane	15·05
Nonane	15·0
Decane	15·0
Naphthene Series—	
Cyclohexane	14·7
Hexahydrotoluene	14·7
Hexahydroxylene	14·7
Aromatic Series—	
Benzene	13·2
Toluene	13·4
Xylene	13·6
Alcohol Series—	
Ethyl Alcohol	8·95
Methyl Alcohol	6·44

It is very interesting to note how slight a variation there is in the thermal efficiency obtainable from the large range of fuels available, but this, at a compression ratio of 5 to 1, is found to lie between 31·4 per cent and 32 per cent for the whole range of petrols, while for the heavy aromatics it is 27·6 per cent and for the alcohols about 32·5 per cent. In the case of kerosene the

thermal efficiency obtained is lower owing to deposition of some portion of the fuel on the interior of the engine which escapes combustion.

Boiling Point. It is clear that a low boiling point in the fuel is desirable, because the higher it is, the greater is the liability of condensation on the cylinder walls, which may result in the leakage of petrol into the crank-case and dilution of the lubricating oil. It must be remembered, in this connection, that all commercial petrols are mixtures of paraffins with naphthenes or aromatics, and as these all have different boiling points, some fractions of the fuel may, and will, condense before others. Generally, however, there is no special danger from this so long as the final boiling point does not exceed 400° F., and ordinary petrols conform to this requirement. If, however, the engine is intended to run on kerosene, the high temperature of boiling (and therefore of condensation) means that some deposition in the cylinder must occur, and as the cylinder walls are not sufficiently hot to counteract this, crankcase dilution is almost certain to happen, and may be very troublesome.

Starting. It is always necessary when starting from cold, to provide for a "rich mixture," which will be dealt with under the heading of carburation, and in the case of composite fuels like petrol, some low boiling point fractions must be present, and their presence is of greater importance than the *mean* boiling point, hence the provision nowadays of special petrol for winter starting.

In the case of alcohol, which is not a mixture but a homogeneous fuel, there are no low boiling fractions, and starting may be so difficult that a mixture of some fuel, such as ether, may be essential until the engine is warmed up.

Calorific Value. The calorific value of a fuel is a measure of the heat developed during its combustion,

and can be ascertained by burning it in some form of calorimeter, but this is complicated by the fact that part of the products of combustion consist of water vapour which has a high latent heat, and as this cannot be made use of in the engine, it is usual to deduct the heat due to the condensation of the water vapour formed, from the total heat produced. The calorific value found after making this correction is termed the "lower calorific value," and this is taken as the basis on which to calculate the thermal efficiency of the engine. As the latent heat of the water vapour has to be *deducted* from the total, it follows that the latent heat of the liquid fuel should be added to the calorimeter results, since this latent heat is actually used in the engine when evaporation is complete before combustion starts. (Note: This is not the case in a Diesel engine.) The calorific value of a fuel therefore determines the quantity of fuel required, the higher the heat value the less being the quantity needed to perform the same work, but *the calorific value does not determine the power output obtainable.*

The following table gives a general indication of the lower calorific value of various fuels, which includes the latent heat of the fuel taken at constant volume—

TABLE VII

LOWER CALORIFIC VALUE OF FUELS

FUEL	B.Th.U. per lb. of Fuel
Aromatic Free Petrols	From 18,500–19,250
Kerosene	,, 19,000
Paraffin Series	,, 19,400–19,700
Naphthene Series	,, 18,890–18,940
Aromatic Series (Benzene)	,, 17,460–17,930
Cracked Spirit	,, 18,540
Alcohol Series	,, 10,000–11,800

Heat Value of Mixture. We must remember that no fuel can be used without a large admixture of air to supply the necessary oxygen for its combustion, so in actual practice we are not concerned with the heat value of the fuel base itself, but with that of the actual mixture as we employ it in the engine, and it is on this that the power output depends. It is found that with "correct mixture," that is, the correct proportion of air for complete combustion, all the hydrocarbon fuels furnish within narrow limits the same calorific value per cubic inch of correct mixture. We know also, from what was said in Section 1 (page 21), that the specific volume changes after the chemical change of combustion, and when allowance is made for this, the difference between the heat values of different air-fuel mixtures becomes even less, and it has been ascertained that, taking the whole range of fuels, the total energy liberated by combustion per cubic inch of correct mixture lies between 48 and 49 ft.-lb. Cracked spirit gives very slightly higher results (49·5 ft.-lb) and the alcohols slightly lower (47 to 48 ft.-lb.).

The heat value of the correct mixture is generally termed the "total internal energy" as distinguished from the calorific value of the fuel, which of itself bears no relation to the power output.

Thermal Efficiency of Fuels. The thermal efficiency obtained at any given compression ratio is substantially the same for all hydrocarbon fuels, irrespective of their chemical composition, but exception must be made in the case of the alcohols which exhibit a slightly higher thermal efficiency due to their higher latent heat, and their lower flame temperature, which reduce both the mean and the maximum temperatures throughout the working cycle.

It is further found that the maximum thermal efficiency is obtained with an *excess* of oxygen, i.e.

when the mixture is somewhere about 15 per cent on the weak side, but combustion will be slow and incomplete if this dilution is carried further, although from theoretical considerations the efficiency should increase

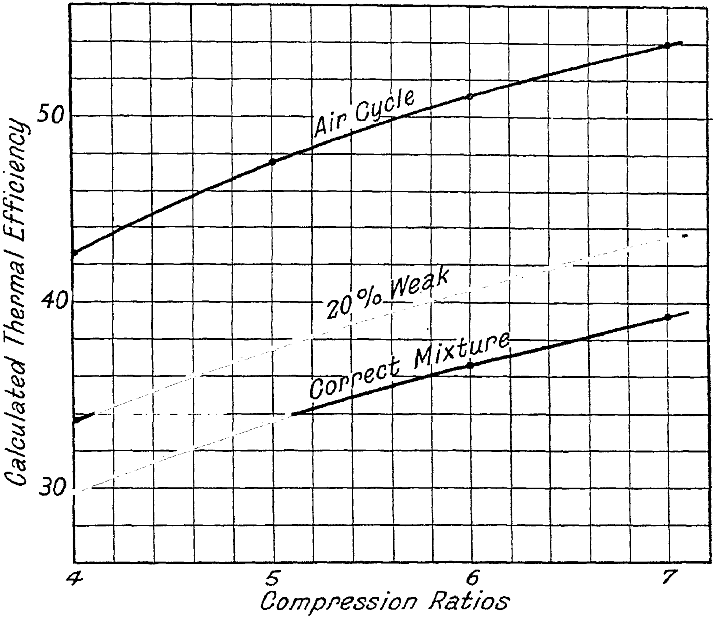


FIG. 2

continuously until pure air, and consequently the air standard, is reached, but naturally this is impossible. In Fig. 2 we have a series of curves showing the calculated efficiency to be obtained at various compression ratios for different mixtures of petrol and air, which shows how the efficiency is increased as the mixture is weakened by excess of air and oxygen; the 20 per cent weak mixture approaching appreciably nearer the

air cycle efficiency than is the case with a correct mixture.

In the following table are given figures showing the lowest fuel consumption in pounds and in pints per indicated horse-power per hour, for various fuels at their highest useful compression as determined from Ricardo's experiments—

TABLE VIII

FUEL CONSUMPTION PER INDICATED HORSE-POWER

FUEL	Consumption per i.h.p. per Hour at h.u.c.	
Aromatic Free Petrol . .	·389–·457 lb.	·405–·503 pints
Paraffin Series . . .	·405–·491 lb.	·473–·568 pints
Naphthene Series . .	·385–·429 lb.	·392–·461 pints
Aromatic Series . . .	·381–·392 lb.	·354–·355 pints
Cracked Spirit . . .	·405 lb.	·428 pints

Maximum Power Output. We have already noticed that the maximum power output varies with the internal energy of the fuel and its latent heat of evaporation, and while the former varies very little as between different fuels, the latent heat, as given in Table V, covers a wide range of values, but in practice the effects of these two factors just about balance, so that in the long run very little variation in the power output can be observed. It may be noticed that the performance of vehicles on the road may vary materially with different fuels, but we must not be misled by this, which can usually be attributed to idiosyncracies in design, particularly as regards the form of the combustion head of the cylinders. Such characteristics may seriously interfere with the rate of burning, etc., and upset the results which should be obtainable. It is only by testing the fuels in an engine of known correct design that their true values can be ascertained.

From the foregoing considerations we may conclude that liability to detonation is the most important factor in determining the suitability of a fuel for use in a petrol engine, and detonation usually depends on the rate of burning of the fuel, and the tendency to detonate is less with slow-burning fuels, hence a low rate of burning is always advantageous.

Any fuel which is capable of standing a high ratio of compression will operate equally well in a low compression engine, provided there is sufficient turbulence in the combustion chamber, but naturally, under such conditions, what might be a most valuable characteristic of the fuel is not being taken advantage of.

We shall refer to dissociation later, but may observe here that losses due to dissociation and the variation of specific heat at high temperatures are practically the same in any kind of hydrocarbon fuel.

Apart from the effects of detonation, the power output of various fuels (except alcohol) does not vary more than about 2 per cent if proper precautions are taken.

Experiments have proved that the performance of any mixture of hydrocarbon fuels as regards detonation, is what would be expected from the performance of the average of each of its component substances, and if the constituents of a commercial mixture constituting "petrol" are known, its performance can be predicted, hence mixtures can be prepared to furnish practically any desired characteristic.

The highest useful compression ratio of any petrol depends on the relative proportions of the paraffins, naphthenes, and aromatics in its composition, the less of the former and the more of the latter the better, and consequently the specific gravity of the mixture, as we have already noticed, forms of itself no guide as to its quality.

While a proportion of highly volatile constituents is an advantage in starting an engine from cold, this may be to a great extent offset by reason of mechanical devices—automatic or otherwise—in the carburettor which permit of a rich mixture being employed in any case for starting.

In carrying out tests of fuel, the result so much depends on the compression ratio that a special engine is required in which the compression ratio can be varied as necessary whilst the engine is running at full power, and it must be pointed out that to vary the ratio by altering the form of the combustion space, as for example, by the makeshift device of fitting pistons with more or less convexity, is certain to give misleading results. All conditions during the test, including the shape of the combustion space, must remain constant. In the Ricardo test engine, the compression is varied by moving the entire cylinder up or down with regard to the piston and connecting rod, so that all the conditions remain constant and no change is introduced which could vitiate the results.

Detonation. The phenomenon of detonation presents a somewhat complicated problem. Briefly, it is the difference between "burning" as contrasted with "explosion," and it occurs when the portion of the charge first ignited by the spark burns and expands with such rapidity that the unburnt portion is compressed before it beyond some certain rate. This compression necessarily produces heat, and if this heat is not carried off by conduction, etc., the unburnt portion of the charge ignites spontaneously and almost instantaneously throughout its whole volume, producing an explosion which strikes the cylinder walls with a hammer blow, resulting in a ringing sound familiar to drivers as "pinking." The general pressure and consequently the temperature within the cylinder being

highly raised, any projecting surface which may be imperfectly cooled thus becomes red-hot, and retains such a temperature that subsequent charges of gas are pre-ignited when compression is in progress, and a state of persistent pre-ignition is thus set up. Since the initial detonation of part of the charge depends on the rate of burning, it is important to ascertain what it is that controls this rate; it is now known that turbulence has little influence on the tendency to detonate.

At one time it was thought that detonation was due only to the temperature of compression, but the difference in the temperatures as between compression ratios of, say, 4 to 1 and 6 to 1 is insufficient to account for it; but all experiments show that with any good petrol detonation will not occur at 4 to 1 while it certainly will at 6 to 1, although the difference in the absolute temperatures at these ratios is only about 70° F.

The following table gives the approximate compression temperatures of a correct petrol-air mixture assuming a constant amount of pre-heating and constant latent heat.

TABLE IX

Showing Approximate Temperatures for Different Compression Ratios

Compression Ratio	Temperature at End of Compression ° F. Absolute
3·5–1	717°
4·0–1	740°
4·5–1	760°
5·0–1	780°
5·5–1	795°
6·0–1	812°
6·5–1	825°
7·0–1	835°
7·5–1	840°

A great deal of research work has been done on the subject of detonation, and the results are too extensive to be more than briefly summarized here, but it has been confirmed that detonation depends mainly on the rate of burning of that portion of the charge first ignited—the rate of burning increases very rapidly with slight increase of compression temperature. The maximum flame temperature may be high enough to produce detonation, and the flame temperature varies with the amount of residual exhaust gases present, which dilute the combustible mixture. The flame temperature can also be reduced by weakening the mixture strength, but there is little doubt that the dilution of the gas mixture by exhaust residues, just referred to, has a most important effect—if there is no such dilution detonation is much more likely to happen.

These conclusions point to the fact that the detonation point is determined by the temperature at which self-ignition occurs in the particular mixture, and the rate at which the burning accelerates as this ignition temperature is exceeded; both these factors are mainly dependent upon the chemical composition of the fuel.

Tests have shown that if the exhaust products are removed by some method of scavenging with air, detonation at once becomes more severe, indicating that the presence of a certain amount of burnt gases is a fortuitous advantage. Experiments have also proved that a proportion of exhaust gas drawn with the air through the carburettor allows the compression to be raised—the increased amount of compression depending on the quantity of exhaust gas added.

Anti-detonators. In view of the harmful effect of detonation, determining, as it does, the ratio of compression possible in any engine, it is quite common to add to the hydrocarbon fuel some agent which will delay the rate of burning. Particularly is this the case

where, for special purposes, a very high compression is needed, as in the engines used for such aviation purposes as the Schneider Cup contests, where every expedient has to be resorted to in order to obtain the last fraction of power from an engine of a given weight. Such engines commonly operate on a compression ratio of 10 to 1, at which no ordinary petrol fuel could possibly be employed on account of detonation. Various substances may be employed for the purpose, the most usual thing being a mixture of tetra-ethyl-lead with a small proportion of ethyl-bromide. Very little of this material is required—usually something of the order of 0·04 per cent to 99·96 per cent of petrol. Several such mixtures are commercially available.

Carbon bisulphide may also be employed, and its behaviour in this connection is of interest, for by itself it has an exceedingly low ignition temperature, and cannot be used without pre-ignition at even so low a compression ratio as 3·6 to 1; yet when mixed with petrol it will allow of a higher compression ratio without detonation than is otherwise possible.

For modern aviation engines the high compression used demands special fuel giving high performance without detonation, and leaded spirit is universal. Cracked spirits are little used, as they are unsuitable: an average aviation spirit would contain, say, 50 per cent of mixed paraffins, 30 per cent aromatics, and 20 per cent naphthenes. Recent research has produced on a commercial scale constant boiling-point spirit, consisting of an isolated hydrocarbon, and mixtures of this give iso-octane having an octane number of 96 to 98, pure iso-octane having a value of 100.

In the final spirit this iso-octane is blended with about 50 per cent of special gasoline, and with the addition of tetra-ethyl-lead a fluid is produced with an octane number of 100.

More recently iso-propyl-ether has been substituted for iso-octane, but all these mixtures present technical difficulties of manufacture.

In automobile racing engines a typical fuel mixture would be petrol 30 per cent, benzol 20 per cent, and alcohol 50 per cent.

It is seen, therefore, that the technology of fuels is becoming increasingly complex.

Range of Burning. By the range of burning is understood the possible range of mixture strength of air and petrol which can be employed, and here again there is very little difference in the behaviour of the various fuels. Compared with the possible range of mixture, in the case of air and coal gas, for instance, the petrol-air range is very narrow. Taking the "correct" mixture —on one side we have a rich mixture, i.e. with an excess of petrol; this is, however, of little interest, since in any case over-richness would be uneconomical. With mixtures on the weak side, however, the case is different, and we find more than one reason why a weak mixture is desirable; we must, however, limit the extent of the "weakness" to that at which the fuel will burn completely—a "correct" mixture is that which will satisfy the theoretical considerations of the chemical equations of the process of combustion, but combustion is still, for practical purposes, complete within a tolerably wide margin either way, and more particularly on the weak side, when there can be no doubt as to the necessary amount of oxygen being available. The difficulty that may arise with a weak mixture is that such a gas burns slowly, and by increasing this condition it is possible to delay matters to such an extent that combustion may persist right into the exhaust stroke, and even extend to the point where the gases may still be burning when the inlet valve opens; in that case uncompressed gas is ignited in the induction

pipe and flame strikes back into the carburettor itself, causing the well-known action of back-firing. It may be stated that so long as the fuel will burn completely, the weaker it is, the lower is the flame temperature. To some extent the danger of back-firing can be prevented by advancing the ignition and so ensuring that combustion commences at the earliest possible moment. The lowest limit of weakness of the mixture for practical purposes lies between 10 and 20 per cent below the correct mixture. From 12 to 18 per cent weak, the power loss from delayed or incomplete combustion is about balanced by the gain in efficiency due to the lower flame temperature, and from 18 to 20 per cent weak is about the limit in an ordinary engine, beyond which back-firing is sure to arise, and experiments all confirm that the maximum efficiency is reached between 10 and 18 per cent weak, depending on the design of the engine.

In a single cylinder engine the mixture can be closely controlled, but in a multi-cylinder engine, on account of the relatively long induction passage to at least some of the cylinders, it becomes practically impossible to supply a mixture of the same strength to all the cylinders, hence if we aim at 15 per cent weak mixture, some of the cylinders may be receiving it as low as 20 per cent, or even weaker, leading to irregular running, loss of power, and possibly back-firing. In practice, therefore, it is best to aim at a mixture containing about 10 per cent excess of air, and as the "correct" mixture for petrol is, say, 15 to 1, this will mean endeavouring to maintain a 16·5 to 1 proportion. This will even things out, but in any case, owing to the variation of the mixture between the cylinders, the indicated thermal efficiency of an engine will be lower as the number of cylinders is increased.

The following table illustrates the effect on the

thermal efficiency and the mean effective pressure by altering the mixture strength with the ignition timing fixed—

TABLE X

THERMAL EFFICIENCY AND M.E.P. FOR VARYING MIXTURES

Air-Petrol Mixture Strength	Thermal Efficiency	M.E.P.
18% weak	30 %	110 lb. per sq. in.
15% ,,	31 %	114 ,, ,,
10% ,,	32 %	120 ,, ,,
5% ,,	31·7%	126 ,, ,,
Correct	31 %	130·5 ,, ,,
5% rich	30 %	134 ,, ,,
10% ,,	28·6%	136·5 ,, ,,
15% ,,	27·2%	138 ,, ,,
20% ,,	26 %	139 ,, ,,
25% ,,	24·5%	138·5 ,, ,,
30% ,,	23 %	137 ,, ,,

If alcohol is used as fuel the results are much the same, except that the drop in thermal efficiency is less with rich mixtures and, in consequence, the mean effective pressure continues to increase up to about 40 per cent rich. The behaviour of benzol in this respect corresponds closely to that of petrol, and in fact, so far as weak mixtures are concerned, there is no practical difference between the three classes of fuel. It will be noticed from the table that the efficiency is at a maximum when the excess of air is about 10 per cent; at 20 per cent weak the combustion is slowed down to the point where back-firing takes place owing to the opening of the inlet valve.

We can therefore conclude that—

1. The mixture range possible is a narrow one with any petrol fuel.

2. Multi-cylinder engines, being unable to obtain

uniform mixture strengths in all cylinders, must always have a somewhat higher fuel consumption than a single cylinder engine, which can only be corrected by increasing the number of carburettors.

3. With all fuels a slight increase of power (but not of thermal efficiency) is obtained with a small excess of fuel, a characteristic which is most marked in the case of alcohol.

4. In single cylinder engines, maximum economy is obtained at about 93 per cent of full power, while in multi-cylinder engines the maximum economy is slightly lower, and is obtained at about 96 to 97 per cent of the full power.

Temperatures During the Cycle. It will be appreciated that the gas temperatures, which change so materially with each episode of the complete cycle, are of great importance in studying the behaviour of any engine. It will be remembered that in Section I, in calculating the pressures and temperatures, the cycle was considered to commence at the beginning of compression; the reason for this soon became apparent, seeing that, other things being constant, the temperature at that point determines the rest of the temperatures throughout the cycle.

The temperature at the same point is important for another reason—it determines, to a great extent, the volumetric efficiency of the engine; at least, other things remaining constant, the volumetric efficiency will vary inversely with the temperature. We have already seen how the temperature is affected by the latent heat of the fuel, and the more the incoming gases are heated and rarified the less will be the weight of fuel drawn into the cylinder.

During the suction stroke, too, the temperature is influenced by the temperature of the exhaust, as there must always be hot residual gases in the combustion

chamber at the end of the exhaust stroke with which the fresh charge has to be mixed and thereby heated. It is usual to carry out the calculation by *assuming* a residual exhaust temperature, and working out the cycle of temperatures from that starting point. Actually a considerable change in exhaust temperature is needed to affect very materially the suction temperature, so there is usually a fair margin for any error in the initial assumption which can be corrected after calculating round the cycle.

We will assume the case of a cylinder of 80 cub. in. capacity (swept volume), and a compression ratio of 5 to 1. Then the clearance space will occupy 20 cub. in., making the total cylinder volume 100 cub. in. Taking normal conditions, we may further assume the following—

R.p.m.	2,000
Mean jacket temperature . .	140° F.
Heat added to the charge by preliminary heating . . .	0·05 B.Th.U. per cycle
Absolute cylinder pressure at end of exhaust stroke	14·7 lb. per sq. in.
Absolute cylinder pressure at end of suction stroke	14·0 lb. per sq. in.
Atmospheric temperature . .	60° F.
Fuel	Petrol containig 50% paraffins, 35% naphthenes, and 15% aromatics.

Such a fuel will possess the following characteristics—

Specific gravity	·740
Boiling point	160°–400° F.
Calorific value . . .	19,000 B.Th.U. per lb.
Latent heat of evaporation .	135 B.Th.U. per lb.
Energy per cub. in. of mixture	46·2 ft.-lb.
Correct mixture ratio . .	14·3 to 1
Change in specific volume after burning	+ 5%

Starting in this case at the commencement of the *suction* stroke, the clearance space then contains 20 cub. in. of hot exhaust gases at atmospheric pressure;

they will probably be at a temperature of about 2100° F. abs. The volume of these gases will be $20 \times 491/2100 = 4{\cdot}68$ cub. in. *at normal temperature and pressure.* The charge drawn into the cylinder consists of air at a temperature of 60° F. and a small quantity of petrol probably vaporized by heat it has taken up from the air; the mixture ratio is 14·3, so the evaporation of 1 lb. of petrol with a latent heat of 135 B.Th.U. is accomplished by the heat in 14·3 lb. of air with specific heat at constant pressure 0·237; it follows that the temperature of the air will fall $135/14{\cdot}3 \times {\cdot}237 = 40°$ F.

We may take it that for the volume of air in the cylinder a change in temperature of 1° F. will be produced by ·00067 B.Th.U., hence, in evaporating, the latent heat of the fuel will absorb $40 \times {\cdot}00067 = {\cdot}00268$ B.Th.U. The cylinder walls are at the jacket temperature of 140° and the valves and piston are hotter still, and from these the gases will probably absorb about ·0005 B.Th.U. per cubic inch or 0·04 B.Th.U. per cycle; the gain of sensible heat in the charge of gas will be sufficient to raise its temperature to about 155° F., assuming all the fuel is evaporated

We have, therefore, 80 cub. in. of fresh charge at 155° F. or 614° F. abs., and at 14·0 lb. per sq. in. absolute pressure, and reduced to normal temperature and pressure this becomes

$$80 \times \frac{14{\cdot}0}{14{\cdot}7} \times \frac{491}{614} = 60{\cdot}9 \text{ cub. in.}$$

and $\dfrac{60{\cdot}9}{80} = 76{\cdot}2$ per cent, which is the volumetric efficiency, and careful tests carried out at the specified temperatures have confirmed this figure.

We have seen that the volume of residual gases was 4·68 cub. in. so that the volume of the combined

mixture of new and exhaust gases is $60 \cdot 9 + 4 \cdot 68 = 65 \cdot 58$ cub. in., and as this fills a volume of 100 cub. in. at a pressure of 14·0 lb. per sq. in. its temperature will be

$$\frac{100}{65 \cdot 58} \times \frac{14 \cdot 0}{14 \cdot 7} \times 491 = 717^\circ \text{ F. abs.}$$

The sources of error in this determination are the assumed temperature of the exhaust products, the heat picked up from the cylinder walls, and the amount of pre-heating. Ricardo, from a number of tests with proper precautions, has found that the figure as calculated, is correct within 10 per cent either way. With benzene as fuel, owing to its higher latent heat and the lesser quantity of air required, the final suction temperature is lower, and the volumetric efficiency under the same conditions would be increased to about 78·5 per cent, while with alcohol the very high latent heat, and still smaller amount of air, so reduce the suction temperature that the volumetric efficiency will be well over 80 per cent. We may, therefore, safely state that using petrol fuel at correct mixture the suction temperature will be 258° F. and the volumetric efficiency 76·2 per cent, while with a 20 per cent weak mixture the suction temperature will be 263° F. and the volumetric efficiency 75·5 per cent; the difference in the suction temperatures being due to the lesser amount of latent heat in the smaller quantity of fuel which will serve to reduce the air temperature.

Compression Temperature. During compression we have assumed that the fuel-air mixture is compressed adiabatically to one-fifth of its initial volume; at the commencement of compression the mixture is below the temperature of the cylinder walls and the gases will absorb heat, while later in the stroke they will lose heat, and it is found that to correct this an alteration

is required in the value of γ, so that instead of following the true adiabatic law of $PV^{1\cdot4} =$ a constant, the exponent for petrol or benzene will become at 2,000 r.p.m. 1·35, and the curve can be calculated from $PV^{1\cdot35}$ = a constant.

In the case of alcohol the exponent may be taken as 1·33. We can then obtain the absolute temperature at the end of compression by multiplying the final absolute suction temperature by $5^{1\cdot35-1}$; from the equation $T_1 = 5^{1\cdot35-1} \times T$ or $T_1 = 5^{\cdot35} \times 258° + 459$ which latter is the suction temperature for petrol; $825 + 459 = 717°$ F. abs. Then $\log 5 = \cdot6989$; $\cdot6989 \times \cdot35 = \cdot2446 = \log 1\cdot755$ and $1\cdot755 \times 717 = 1258°$ F. abs., the temperature after compression for the correct mixture. Similarly, we find that for a 20 per cent weak mixture the compression temperature will be 1,267° F. abs.

In the case of benzol, the temperatures after compression will be 1,220° F. abs. for correct mixture and 1,232° F. abs. for 20 per cent weak mixture.

Combustion Temperature. Combustion in a petrol engine takes place at constant volume, hence the whole of the energy stored in the mixture is available for increasing its internal energy, except so much as is lost by conduction to the walls of the cylinder; there are however, factors which render the calculation of temperatures very complicated, as apart from the loss by conduction, there is the change in specific heat of the gas which *increases* with the temperature and also the phenomenon next dealt with.

Dissociation. At the high temperatures prevailing, the products of combustion, CO_2 and water, split up or dissociate into carbon monoxide, oxygen, and hydrogen, as shown by the formulae

$$2CO_2 = 2CO + O_2$$
$$2H_2O = 2H + O_2$$

When a correct mixture of benzene and air is burned the combustion will take place according to the following chemical equation

$$2C_6H_6+15(O_2)+(56{\cdot}4N_2)=12CO_2+6H_2O+56{\cdot}4N_2$$

and from the first half of the equation it is seen that the number of molecules before combustion is $2 + 15 + 56{\cdot}4 = 73{\cdot}4$. After combustion the number becomes $12 + 6 + 56{\cdot}4 = 74{\cdot}4$. The calculation of the energy absorbed is a long and somewhat complex process and need not be followed out here, but it is known that the specific heat of the products of combustion changes seriously above 1,500° C. or 2,732° F., and dissociation of CO_2 and H_2O occurs to a greater or less extent at high temperatures, and as the maximum temperature of the gas is somewhere about 2,500° C. or 4,532° F., the variation in the specific heat of CO_2 and H_2O becomes marked. The dissociation of these gases is accompanied by an increase in the number of molecules, and as both compounds have one product of dissociation (oxygen) in common, dissociation of a mixture of the two is not the same as when they are separate. It follows, then, that the apparent specific heat of *mixtures* of CO_2 and H_2O, as well as of the other gases present after explosion, may be very different from the sum of the apparent specific heats of each measured separately.

It is due to dissociation that the power varies with the mixture strength; if no dissociation is allowed for, the calculated power is a maximum at the correct mixture for complete combustion, falling off for both richer and weaker mixtures. The calculated maximum thermal efficiency for the correct mixture is 33·8 per cent when dissociation is allowed for, and the following table shows how this varies with and without dissociation for different mixture strengths, and with a compression ratio of 5 to 1.

TABLE XI

EFFECT OF DISSOCIATION ON THERMAL EFFICIENCY

Conditions	Explosion Temperature	Efficiency
Correct Mixture, no Dissociation.	5,473° F.	35·9%
Correct Mixture, with Dissociation	4,811° F.	33·8%
20% Weak, no Dissociation .	4,840° F.	37·9%
20% Weak, with Dissociation .	4,478° F.	37·9%

The strength of the mixture naturally has an important effect on the composition of the exhaust gases, and here also the effects of dissociation become apparent. In Table XII, on page 80, the approximate analysis of exhaust gases is given for various mixture strengths.

When dissociation occurs the maximum temperatures given in the table are not reached, hence *dissociation tends to lower the maximum temperature.*

The equation given above for combustion without dissociation, viz—

$$2C_6H_6 + 15O_2 + 56{\cdot}4N_2 = 12CO_2 + 6H_2O +$$

must, when dissociation is allowed for, be written-

$$2C_6H_6 + 15O_2 + 56{\cdot}4N_2 = a\,.\,CO_2 + b\,.\,CO + c\,.\,H_2O + d\,.\,H_2 + e\,.\,O_2 + 56{\cdot}4\,N_2$$

and the value of the quantities a, b, c, d, and e must be determined by calculation. When the effect of dissociation is taken into account the value given in Table XIII will take the place of those given in Table XII.

Particular attention is drawn to the change in specific volume in these circumstances.

We cannot pursue further the investigation of this subject of dissociation as space forbids it, but it remains to notice an important conclusion. We have seen that

TABLE XII

Composition of Exhaust Gases at Different Mixture Strengths. Without Dissociation

	Mixture Strengths			
	20% Weak	Correct	10% Rich	20% Rich
N_2	56·4	—	56·4	56·4
CO_2	9·6	12·0	10·34	8·71
CO	—	—	2·9	5·69
H_2O	4·8	6·0	6·5	6·89
H_2	—	—	0·1	0·31
O_2	3·0	—	—	—
Volume before Combustion	73	73·4	73·6	73·8
Volume after Combustion	73·8	74·4	76·2	78·0
Explosion Temperature, ° F.	4,840	5,473	5,300	5,135
Compression Ratio	5–1	5–1	5–1	5–1

TABLE XIII

Composition of Exhaust Gases at Different Mixture Strengths. With Dissociation

	Mixture Strengths			
	20% Weak	Correct	10% Rich	20% Rich
N_2	56·4	56·4	56·4	56·4
CO_2	7·70	8·22	7·93	7·46
CO	1·9	3·78	5·27	6·94
H_2O	4·7	5·8	6·31	6·74
H_2	0·1	0·2	0·29	0·46
O_2	4·0	1·99	1·28	0·70
Volume before Combustion	73·0	73·4	73·6	73·8
Volume after Combustion	74·8	76·4	77·5	78·7
Explosion Temperature, ° F.	4,478	4,811	4,874	4,894
Compression Ratio	5–1	5–1	5–1	5–1

the air standard thermal efficiency of an engine is expressed by the following equation: $E = 1 - \left(\frac{1}{r}\right)^{\gamma - 1}$ where gamma is the ratio of the specific heat of air at constant pressure to that at constant volume. Sufficient has been said to show that owing to the variation in these specific heats due to high temperatures and dissociation, the value of γ is a fictitious one; hence the air standard efficiency becomes an unattainable ideal, and therefore is not a fair standard of comparison for a practicable engine. Tizard and Pye, therefore, after a full investigation of all the conditions actually obtaining, and after making all the appropriate corrections for the chemical and physical changes which occur in practice, proposed a new value for gamma, viz., 1·296, from which the equation for efficiency becomes

$$E = 1 - \left(\frac{1}{r}\right)^{\cdot 296}$$

which gives a much lower series of values for practical comparison. The following table shows the new values calculated by the author from the above equation compared with the air standard values.

We thus see that the variation in the specific heat, together with the temperature, alters the value for γ of 1·4, which is its value for ideal air, and a modified exponent is required to compensate for the known or assumed differences between ideal air and actual petrol air mixture. It is appropriate here to examine the difference brought about in the pressure-temperature calculations by this altered value of the exponent, and it will be sufficient to take examples of pressure calculations at compression ratios of 4·5 to 1, 5 to 1, and 5·5 to 1 with the following values of the exponent 1·40, 1·35, 1·30, and 1·25.

Starting with the familiar 5 to 1 ratio and an initial

TABLE XIV

ENGINE EFFICIENCIES FOR VARIOUS COMPRESSION RATIOS (r)

r	$E = 1 - \left(\frac{1}{r}\right)^{\cdot 4}$	$E = 1 - \left(\frac{1}{r}\right)^{\cdot 296}$	r	$E = 1 - \left(\frac{1}{r}\right)^{\cdot 4}$	$E = 1 - \left(\frac{1}{r}\right)^{\cdot 296}$
3·8	0·4138	0·3261	5·8	0·5050	0·4057
3·9	0·4201	0·3316	5·9	0·5084	0·4087
4·0	0·4256	0·3366	6·0	0·5116	0·4116
4·1	0·4313	0·3410	6·1	0·5149	0·4145
4·2	0·4367	0·3461	6·2	0·5180	0·4175
4·3	0·4420	0·3506	6·3	0·5211	0·4200
4·4	0·4471	0·3552	6·4	0·5238	0·4229
4·5	0·4521	0·3595	6·5	0·5270	0·4251
4·6	0·4569	0·3637	6·6	0·5300	0·4280
4·7	0·4615	0·3674	6·7	0·5328	0·4307
4·8	0·4660	0·3717	6·8	0·5355	0·4329
4·9	0·4704	0·3753	6·9	0·5382	0·4354
5·0	0·4747	0·3790	7·0	0·5398	0·4376
5·1	0·4788	0·3826	7·5	0·5534	0·4495
5·2	0·4829	0·3861	8·0	0·5647	0·4596
5·3	0·4868	0·3896	9·0	0·5740	
5·4	0·4906	0·3930	10·0	0·6020	
5·5	0·4944	0·3963	11·0	0·6170	
5·6	0·4980	0·3995	12·0	0·6290	
5·7	0·4984	0·4026			

pressure of $P = 14{\cdot}7$, the final pressure P_1 will, of course, be $P_1 = P \times r^{\gamma}$ where γ has the above values

$$\log 5 = \cdot 6989$$

$$\begin{aligned} \cdot 6989 \times 1{\cdot}40 &= \cdot 97846 = \log 9{\cdot}52 \\ \text{,,} \times 1{\cdot}35 &= \cdot 9435 = \text{,,}\ 8{\cdot}77 \\ \text{,,} \times 1{\cdot}30 &= \cdot 9085 = \text{,,}\ 8{\cdot}10 \\ \text{,,} \times 1{\cdot}25 &= \cdot 8736 = \text{,,}\ 7{\cdot}47 \end{aligned}$$

Multiplying the figures in the last column by 14·75 we obtain the following values of

$P_1 = 140, 129, 119, 109{\cdot}8$ lb. per sq. in.

Proceeding in the same way for 5·5 to 1 compression ratio we find the final pressure calculated on the four exponents to be

$P_1 = 160{\cdot}2, 146{\cdot}7, 134{\cdot}8,$ and $123{\cdot}3$ lb. per sq. in.

and again at 4·5 to 1 compression ratio they become

$$P_1 \doteq 120{\cdot}5,\ 112,\ 104, \text{ and } 96{\cdot}2 \text{ lb. per sq. in.}$$

In Fig. 3 these values are plotted as three curves, which are useful for comparative purposes, and in experimental work such manipulations of the value of γ are frequently helpful in forming conclusions from observed results. From these curves the values resulting from any value of γ between the limits of value of 1·25 and 1·40 can be seen at a glance. In Fig. 4 the same values for the final compression pressures are plotted on the three compression ratios for the four values of γ from which corresponding pressures can be obtained for any intermediate compression ratio between 4·5 to 1 and 5·5 to 1.

TABLE XV

Showing Final Compression Pressures for Different Compression Ratios, and Varying Values of γ, with Initial Pressure of 14·7 lb. per sq. in.

Values of γ	Compression Ratios		
	4·5 to 1	5·0 to 1	5·5 to 1
1·40	120·5	140	160
1·35	112	129	146·7
1·30	104	119	134·8
1·25	96·2	109·8	123·3

Leaving, then, the further calculations on the subject of dissociation we can summarize the conclusions, arrived at by direct experiment, by Ricardo—

Energy content of correct mixture	46·2 ft.-lb. per cub. in.
Total internal energy .	48·5 ft.-lb. per cub. in.
Compression ratio . .	5 to 1
Indicated thermal efficiency	31%
Volumetric efficiency .	76·2%
Maximum flame temperature	4,487° F.
Maximum flame temperature absolute	4,946° F. absolute
Corresponding energy content . .	44·5 ft.-lb. per cub. in.

Heat drop due to change in spec. vol. .	13·6 ft.-lb. per cub. in.
Compression work restored in expansion	3·6 ft.-lb. per cub. in.
Cylinder wall loss during expansion .	2·8 ft.-lb. per cub. in.
Total heat drop during expansion $= 13\cdot6 + 3\cdot6 + 2\cdot8$ . . .	20 ft.-lb. per cub. in.
Final energy content, $44\cdot5 - 20$.	24·5 ft.-lb. per cub. in.
Corresponding final temperature .	3,047° F.

After the expansion stroke the gas temperature is thus about 3,050° F., or 3,509° F. abs., at a pressure of about 70 lb. per sq. in. At the point of exhaust the gas will expand rapidly to atmospheric pressure and its temperature will fall in the ratio $\left(\frac{14\cdot7}{70}\right)^{\frac{\gamma-1}{\gamma}}$ where γ is 1·30. This brings the temperature down to 2,450° F. abs., and further heat loss during the exhaust stroke will further reduce this to about 2,100° F. abs., which is the temperature we first assumed. A theoretical indicator diagram from an engine operating under the above conditions would, at the four corners, exhibit the following pressures and temperatures—

1. Commencement of compression	$P = 14\cdot0$ lb. per sq. in. $T = 717°$ F. abs.
2. After compression	$P = 123$ lb. per sq. in. $T = 1{,}258°$ F. abs.
3. Before expansion	$P = 508\cdot5$ lb. per sq. in. $T = 4{,}946°$ F. abs.
4. After expansion	$P = 72\cdot2$ lb. per sq. in. $T = 3{,}509°$ F.

and the mean effective pressure would be 137·4 lb. per sq. in. with a compression ratio of 5 to 1.

The mean effective pressure is arrived at by multiplying together the thermal efficiency, the total internal energy (ft.-lb. per cub. in.) and the volumetric

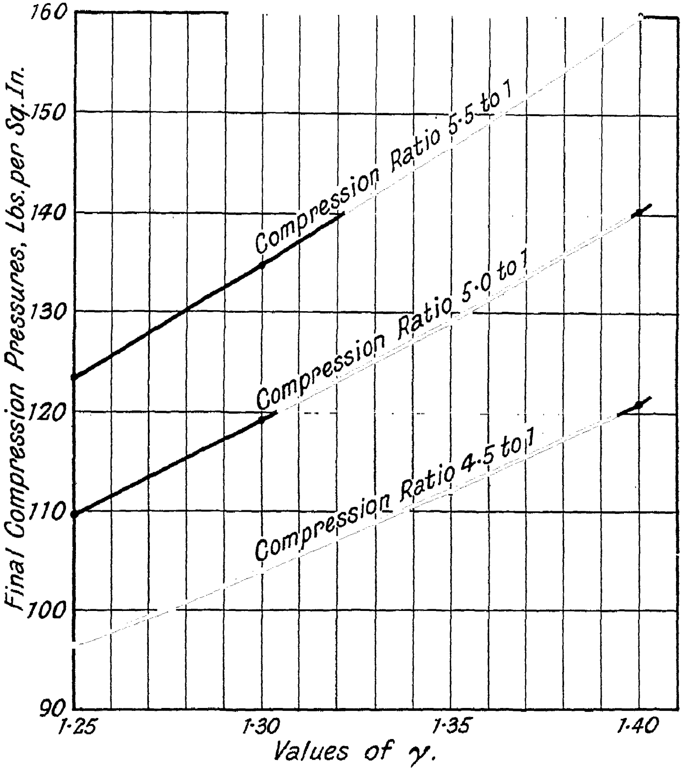


FIG. 3. FINAL COMPRESSION PRESSURES

For varying values of gamma at different compression ratios starting from initial pressure of 14·7 lb. per square inch

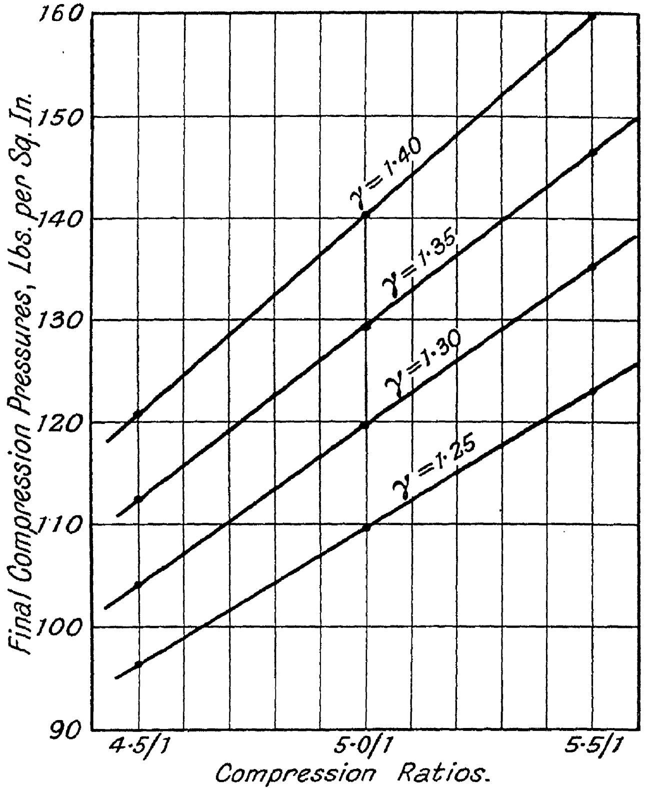


FIG. 4. FINAL COMPRESSION PRESSURES

For varying compression ratios, for different values of gamma starting from 14·7 lb. per square inch

efficiency × 12. It can, of course, also be determined graphically from the diagram. In the above case

$$\cdot31 \times 48\cdot5 \times 12 \times \cdot762 = 137\cdot4$$

The change in specific volume after combustion affects the energy liberated per cubic inch of mixture, and to obtain the true value we must multiply the energy liberated per cubic inch by the change in specific volume. Thus, for petrol, the heat energy per cubic inch is 46·2 ft.-lb. and the change in specific volume is + 5 per cent, so 46·2 + 5 per cent = 48·5 ft.-lb., the total internal energy per cubic inch.

For benzine the total internal energy is 47·5, and for alcohol 47·4 ft.-lb. per cub. in.

If we are working with a weak mixture, the total internal energy is, of course, reduced, but owing to the lower flame temperature the engine will operate at a higher efficiency; the latent heat of evaporation is also quantitatively somewhat lower, which to some extent tends to reduce the volumetric efficiency. We saw that for a correct mixture the indicated mean effective pressure was 137·4 lb. per sq. in., so with a 20 per cent weak mixture this figure will be reduced to 118 lb. per sq. in.

If a mixture 20 per cent *rich* is taken, there will be a slight gain in *power* (about 4·5 per cent) as the specific volume after burning will be slightly greater, and the latent heat of the greater quantity of petrol will increase the volumetric efficiency, and the thermal efficiency is also slightly higher.

Hence with

a mixture 20 per cent weak	I.M.E.P.	= 118 lb. per sq. in.	
correct mixture	,,	= 137·4	,,
20 per cent rich mixture	,,	= 143·58	,,

With corresponding variations in the horse-power obtainable.

Heat Distribution in a Four-cycle Engine. Having now examined the conditions obtaining within the engine which result directly from the processes of combustion of different mixtures of fuel, it remains to trace out how the heat produced is distributed and used throughout the cycle, and how large a proportion of it is rejected finally to the exhaust. Not only is such a study important in making the preliminary calculations for the design of any engine (and incidentally of its radiator), but a correct knowledge of this is essential if experiments on actual engines, and the regular testing of them, are to be intelligently carried out and interpreted.

Generally, the cycle of changes through which the heat passes is expressed in terms of, *first*, the proportion converted into indicated horse-power; *secondly*, the proportion lost to the cylinder walls and carried away by the cooling water; *thirdly*, the proportion rejected to the exhaust.

This latter is what remains after deducting the first two items from the total heat supplied by the fuel; as a rule the losses by radiation have to be included in the third category as they cannot be directly measured. It must, however, be emphasized that the foregoing subdivision is only a convenient and arbitrary form of expressing the heat distribution, and hence must always be subject to reservations. It happens that there is no other way of measuring the heat, but various corrections have, in practice, to be made if the results are not to be entirely misleading.

The proportion of the total heat of the fuel which is converted into indicated horse-power can be quite easily and correctly determined by direct measurement of the known horse-power produced, and the heat which is carried away by the cooling water from the cylinder walls, the valves, and the combustion head can be measured with tolerable accuracy by multiplying the

weight of water circulated by the difference between its incoming and outgoing temperatures, but we must remember that the jacket water carries away also the heat given up by radiation, conduction, and convection, which is generated during the period of combustion. During the expansion period also, the parts of the engine actually in contact with the heated gases are being cooled by the water, and this action cannot be arrested (however desirable that might be) while expansion is in progress; lastly, during the exhaust stroke the same cooling action continues, so that if we were to measure the heat in the exhaust *gases* this does not include *all* the exhaust heat, some of which is being carried off in the cooling water. It is evident that the correct estimate of the subdivision of the heat distribution is not quite so simple as it first appears. We will examine separately (1) heat given up by radiation, conduction, etc., during the period of combustion; (2) heat abstracted during expansion; and (3) heat given up during exhaust.

Heat Lost During Combustion Period. Although the period of combustion in a constant volume engine is excessively short, it is, nevertheless, of measurable duration, and, moreover, at that time the gases are at an extremely high temperature, actually between 4,200° F. and 4,500° F. when the volatile fuels we have already considered are employed, so that the temperature difference between the inside and the outside of the combustion head is very great; this period also is one when the gases in the combustion chamber are in a state of violent agitation, so that the heat the gases possess is very readily conveyed to the metal surfaces by conduction and convection. Were it possible by any means to suspend the transfer of heat to the cylinder walls during the combustion period, the heat so conserved could be converted into indicated

horse-power at whatever efficiency was due to the range of expansion alone, exclusive of the negative work performed during the compression period. In an engine operating at a compression ratio of 5 to 1, this expansion efficiency would be about 40 per cent, and the remaining 60 per cent of the heat conserved would fall, to be rejected finally into the exhaust.

Heat Lost During Expansion Period. Naturally, the rate at which heat is lost during the expansion stroke is not constant, owing to the wide variation in the temperature of the gases throughout the period. At the commencement of the stroke the temperature is practically that of combustion, and the loss is as rapid as it is during combustion, and could that loss have been arrested, we could have utilized the heat at an efficiency which would correspond to the whole range of expansion. On the other hand, the heat lost towards the end of the expansion stroke is of less importance, for not only is the temperature much lower, and the consequent heat transfer less, but had we conserved the heat it would have performed but little useful work during the short remainder of the stroke, so that nearly all of it would have had to be rejected into the exhaust at the end of expansion.

Although we should at first suppose from the foregoing reasoning that the heat loss would be much greater during the first part of the expansion stroke, we find that as expansion proceeds, more and more of the relatively cool cylinder walls become exposed, and owing to the process of dissociation and subsequent recombination, the fall in temperature is much less than might be supposed, and at 5 to 1 compression ratio the final temperature is over 3,000° F.

From these considerations, although it is common to add together the heat lost during combustion and expansion as though they were the same, such a course is

inaccurate and misleading; and in the case of the heat lost during expansion, it is unlikely that more than 20 per cent could be converted into useful work, and the remaining 80 per cent would have to be lost in the exhaust.

Heat Lost During Exhaust Stroke. During the exhaust stroke the temperatures are much lower, but at this stage heat is readily taken up by the water, not only from the regular flow of heat through the cylinder walls, but as the gases issue through the exhaust valves at a very high velocity, and through that portion of the exhaust pipe which is within the jacket and therefore cooled by the water. It may be estimated that of the total heat the water carries away not less than one-half is picked up during the exhaust period, and the proportion may be even greater than this; it follows that the whole of the heat taken up during the exhaust stroke, the greater part of that taken up during expansion, and about 60 per cent of that taken up during combustion should not be considered as water-jacket losses at all, but ought by rights to have been added to the *exhaust* losses.

There is also a considerable amount of heat generated by the friction of the piston against the cylinder walls, and this also has to be removed by the cooling water.

We may now take a specific example and ascertain as closely as possible the real gain in efficiency which would result were we able to eliminate all the loss of heat to the cylinder walls. We will assume an engine working under the following conditions—

Compression ratio, 5 to 1.

Heat of the fuel converted into useful work on the pistons, 32 per cent.

Heat carried away by the cooling water, 28 per cent.

$32 + 28 = 60$ per cent of the total heat of the fuel, which leaves 40 per cent of the total heat remaining to be accounted for as lost in exhaust, radiation, etc. We

may further subdivide the 28 per cent of the total heat which is taken up in the cooling water as—

(a) Lost to cylinder walls during combustion period .	6%
(b) Lost to cylinder walls during expansion period .	7%
(c) Taken up during exhaust stroke	15%
TOTAL	28%

Now had we been able to arrest these losses, of the 6 per cent lost during combustion about 40 per cent might have been converted into useful work, and $\frac{40 \times 6}{100} = 2{\cdot}4$ per cent of the total heat of the fuel.

Of the 7 per cent lost during expansion we could perhaps have utilized about 20 per cent, and $\frac{20 \times 7}{100} = 1{\cdot}4$ per cent of the total heat of the fuel.

Of the 15 per cent lost during the exhaust stroke the whole would have been lost.

We thus see that although 28 per cent of the total heat of the fuel has been carried off by the cooling water, only $2{\cdot}4 + 1{\cdot}4 = 3{\cdot}8$ per cent of the fuel could have been conserved to appear as useful work at the piston; so that this would have been increased from 32 to 35·8 per cent, a gain of about 12 per cent on the total. At the same time we should have been faced with another loss, for by suppressing the heat loss to the cylinder walls the resulting temperature of the working charge would have been higher, and as we have seen there would have been an increase in the specific heat and greater dissociation, and the effect of these added together would have further reduced the net gain, and the 32 per cent would have been increased only to 35 per cent, and probably less.

From these considerations we begin to realize that the loss of heat to the cylinder walls is a relatively

small matter, and it is going much too far to assume that the heat carried away by the cooling water is a criterion of that loss. We should probably be not far from the truth in assuming that only about 10 per cent of the heat carried away in the cooling water might have been converted directly into useful work.

A series of tests carried out by Ricardo gave the following information—

TABLE XVI

HEAT DISTRIBUTION IN A CONSTANT VOLUME ENGINE

Alcohol Fuel .	Heat converted to i.h.p. . .	26·95%
	Heat lost to cooling water . .	24·60%
	Heat lost in exhaust and radiation .	48·45%
	TOTAL	100·00%
Petrol Fuel .	Heat to i.h.p.	26·0%
	Heat to water	28·4%
	Heat to exhaust, etc. . . .	45·6%
	TOTAL	100·0%

R.p.m., 975 to 1,700. Piston speed, 1,300 to 2,266 ft. per min.
Compression Ratio, 3·8 to 1

From these tests it is seen that the thermal efficiency of alcohol is higher than that of petrol at the same compression ratio, hence the proportion of heat carried away by the water is less. The thermal efficiency was affected very little by a wide range of speed; it was further ascertained that the heat carried away by the cooling water falls slightly with an increase in speed.

A series of tests carried out at a compression ratio of 5·45 to 1 further showed how small a part the direct heat loss to the walls actually played, variations in this through an appreciable range did not affect the

thermal efficiency perceptibly. The final conclusions then are—

(*a*) That the direct heat loss to the cylinder walls has a very small influence on the performance of an engine, and even if it were possible to eliminate this loss, the gain in useful work and efficiency would only amount to that due to the conversion of about an extra 2·5 per cent of the total heat of the fuel into useful work.

(*b*) If the total heat carried away in the cooling water were diverted, only a very small proportion could be retained for conversion to useful work, and all the remainder would appear in the higher temperature of the exhaust.

(*c*) When running at full throttle, the heat flow to the water is proportional to the speed of the engine.

(*d*) When throttled down, a larger proportion of heat is abstracted by the water before the gases are exhausted.

Influence of Water-jacket Temperature. It is a matter of common observation that engines run better when hot—giving more power, and a higher efficiency when the water temperatures are relatively high. The immediate cause of this is that when the jackets are cold, there is considerable precipitation of fuel, particularly in those parts of the induction system which are surrounded by the jacket water; the distribution is impaired by the precipitation which takes place differently in the various cylinders, the power and efficiency both suffer as a result, and the engine runs irregularly. The commonly-stated explanation that when the jackets are hot there is less heat conducted away from the cylinder walls, cannot be substantiated for the following reason.

If we take the mean temperature of the cylinder walls as being, say, 3,000° F. (probably a low estimate) and water at 140° F., we have a temperature difference of

2,860° F., but if the water is relatively cold, say, at 100° F., we have a temperature difference of 2,900° F., or an increase in the temperature difference of only 1·4 per cent, which is not sufficient to produce any noticeable result.

A much more important effect of higher jacket temperature is to decrease the piston friction as the viscosity of the lubricating oil is reduced thereby.

If (as is frequently done in experimental engines) the temperature of the induction system is controlled by some independent means, so that it is not affected by the cylinder temperature, the difference in power, due to the jacket temperature alone, is much less evident. Under such conditions the variation in power is clearly due to three separate reasons—

(*a*) The variation in loss of heat from the cylinder walls with the change in the temperature difference.

(*b*) The variation in volumetric efficiency with the cylinder temperature.

(*c*) The variation in piston friction with change in viscosity.

First, with regard to the heat loss from the cylinder walls, we have already seen that with a well-designed engine, and particularly with attention paid to the form of combustion chamber, the whole of the heat passing into the water amounts to not more than 12 or 13 per cent of the total heat of the fuel taken in, and were we able to eliminate this loss entirely, the increase in indicated horse-power would be something less than 10 per cent, having regard to dissociation and recombination at the increased temperature which would by such means be obtained. Assuming a mean temperature during combustion of 3,800° F., the *inner* surface of the cylinder walls when the water is at 212° F. (the maximum possible) will be about 300° F., hence the temperature difference will be 3,800° − 300° =

3,500° F. If the water temperature were reduced to 72° (a reduction of 150°) the temperature difference will be increased to about 3,650°, or, say, 4 per cent. It is known that the heat loss is proportional to the temperature difference, so the decrease in the indicated horse-power due to the change will only amount to 4 per cent of 10 per cent, or $\frac{4 \times 10}{100} = 0{\cdot}4$ per cent. It is not likely that even with an inefficiently-designed combustion chamber the decrease could amount to more than about 1 per cent.

Next, with regard to the change in volumetric efficiency. Here we find a more important difference. With the inner surface of the cylinder walls at 300° F., the rise in temperature of the gases due to contact with the working surfaces will amount to about 80° F. With "cold" water at 72° it will be, say, 25° less, or about 55° F., assuming that in each case the change in the temperature rise of the gases as they enter is about one-sixth of that of the cylinder walls, which has been proved by experiment. The mean absolute temperature of the gases after entering the cylinder will be about 240° F., or 700° F. absolute, and the *weight* of the charge drawn in is, as we have seen, directly proportional to the absolute temperature, and the power obtained varies in the same proportion. It follows, therefore, that if the absolute temperature is reduced by 25°, the weight of the fuel drawn in will be increased by the ratio $\frac{700}{675}$, namely 3·8 per cent.

Consequently, by reducing the water temperature from 212° to 72° we *reduce* the power by increased heat losses from 0·4 to 1·0 per cent.

The weight of charge drawn in, and the power consequently produced, will be *increased* by 3·8 per cent, or

a net increase of from 3·8 – ·4 = 3·4 per cent to 3·8 – 1 = 2·8 per cent.

So far, then, we see that the indicated power is definitely *increased* by a reduction in the jacket water temperature. Tests to prove this statement show that the mean effective pressure is increased by reducing the jacket water temperature to the extent shown by the following table—

TABLE XVII

SHOWING VARIATION IN INDICATED M.E.P. WITH JACKET WATER TEMPERATURE

Jacket Water Temperature	Mean Effective Pressure, lb. per sq. in.
212° F.	136.6
200° F.	137
150° F.	138·5
100° F.	140
70° F.	140·6

Lastly, we have the question of piston friction. Here the temperature has a very marked result, but the effects of difference in piston design enter so largely into the matter that it is not possible to deduce any empirical figures. It can be demonstrated that piston friction is dependent, in a very large measure, on the viscosity of the oil used for lubrication, and this changes rapidly with changes of temperature; the viscosity increases as the temperature falls, adding thereby to the friction.

It has been found that the difference between hot and cold cylinder jackets may produce a variation in brake horse-power, amounting to as much as 8 per cent. It can easily be seen that in such circumstances the gain in horse-power from thermal considerations would be much more than offset by the friction loss (8 – 3 = 5 per cent net loss). With very light pistons,

however, and particularly with pistons having a relatively small bearing surface, the friction losses may be no more than about 3 per cent, in which cases the gains and the losses cancel out and there is no resulting change in the net power produced; while in the case of special pistons of the "slipper" type, the friction losses may be so low that the balance is in favour of the thermal gain, and the net result may be a slight gain in power.

To conclude, therefore, if the carburettor temperature is maintained constant, the power output of any engine may increase or decrease with an *increase* in the cooling water temperature—depending on the relative effect of the piston friction.

If the piston friction is high the power will increase.

If the piston friction is low the power will not increase.

The variation in heat losses over wide limits of cooling water temperature is too small to have any material effect on the power.

The change in volumetric efficiency may be appreciable, but its effect is generally insufficient to balance the change in piston friction.

We have seen that it is possible to calculate the temperature at any and all points of the internal combustion engine cycle, provided the composition of the fuel is known, and *on the assumption that no loss of heat takes place throughout.* This last is of necessity an unattainable condition, and long and special research would be required to obtain the data for the necessary corrections. We have also seen that the maximum power to be obtained from a paraffin or an aromatic fuel is not sensibly different so long as comparison is made at such a compression ratio that full ignition advance is possible without detonation in either case. We have further seen that higher compression ratios are possible with aromatics than with paraffins, and on

thermodynamic grounds the compression em should be the highest possible without detonation; with a properly-designed engine, therefore, the aromatic fuels have the advantage. The discussion of th matters has brought out the fact th air

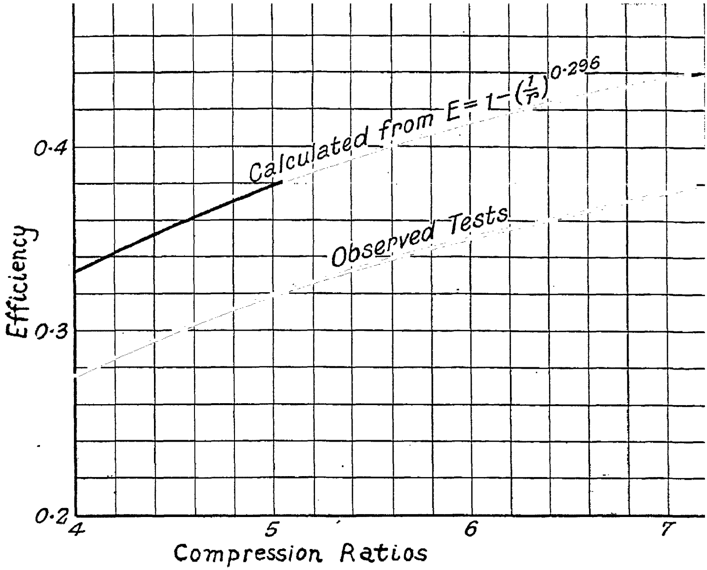


FIG. 5. COMPARISON BETWEEN TIZARD IDEAL EFFICIENCY AND OBSERVED RESULTS OF TESTS

cycle" is an unattainable ideal indicating that a more suitable standard engine of comparison is desirable, because, whatever ideal is adopted, it should be one which will serve as a guide to designers, to furnish a knowledge of what is to be expected from modifications in design when taking into account all the physical and chemical conditions which arise, so that results can be calculated with reasonable facility and independently of any particular engine.

An ideal efficiency for weak mixture strength is to be calculated from the formula $E = 1 - \left(\frac{1}{r}\right)^{\cdot 295}$ and Fig. 5 shows the results to be expected from this as compared with observed thermal efficiencies, from which practical tests are seen to approach the Tizard ideal within 20 per cent.

We have further seen that engines of present-day design will run at their maximum efficiency on a fuel which possesses a toluene value of from 15 to 20 per cent. Detonation is, above all, the feature which limits the value of any fuel, and incidentally limits the compression ratio and hence the efficiency of any engine. Detonation seems to be a direct function of the rate of burning, and this varies with the chemical composition of the fuel. Toluene is the most effective of natural constituents, but when extreme correction is necessary for very high compression ratios, other chemical substances may be employed to secure freedom from detonation.

No engine will start on petrol with an economical mixture strength, and on commercial petrol an engine will only start from cold provided there are sufficient aromatics present to ensure vaporization at comparatively low temperatures.

Perfect distribution in the various cylinders of a multi-cylinder engine is practically unattainable, hence the more cylinders there are the lower will be the thermal efficiency.

From all the considerations it would appear that present-day engines have reached the practical limit of thermal efficiency, and any further development or improvement must be in the direction of providing an engine which will operate on a weaker mixture than is at present possible.

SECTION III

PETROL ENGINE THEORY

BY

A. W. JUDGE, A.R.C.Sc., Wh.Sc., A.M.I.A.E.

SECTION III

PETROL ENGINE THEORY

COMBUSTION AND EXPLOSION

In the section on Thermodynamics it is shown that the modern high speed internal combustion engine may very conveniently be regarded, for theoretical purposes, as a type of hot air engine following a definite sequence, or cycle, of operations, as follows, namely (1) Compression of a given quantity of air. (2) Addition of heat at the end of compression. (3) Expansion of the compressed and heated air, and finally (4) Rejection of part of the heat.

The given quantity of air is thus alternately heated and cooled, and work, or propulsive effort, is exerted on the piston by the expanding air.

Further, it is generally assumed in heat cycles, such as the Carnot, the Otto, and the Diesel (or constant pressure), that the heat is given either instantaneously or slowly as the theoretical considerations require.

In the case of the internal combustion engine the heat which is added in the operation (2), previously mentioned, is actually supplied by the chemical combustion of the fuel contained in the compressed air, or added to this air at, or near, the end of the compression stroke—as is the case in the Diesel-type engine. It is our present object to study this process of fuel burning in more detail, commencing with some elementary considerations of the combustion of simple gases, leading up to the burning of hydro-carbon fuels under similar conditions to those which occur in actual internal combustion engines.

Explosive Mixtures. When an inflammable gas or vapour is mixed with oxygen in certain proportions in a vessel, and a source of ignition, such as a flame or electric spark, is applied, the mixture will combine chemically, or explode with more or less violence. If the proportions of the inflammable gas and oxygen be varied within certain limits it will be found that the sharpness, or intensity, of the resulting explosion will also vary. Moreover, there will be found to be one particular proportion of each gas which gives the most explosive mixture.

The results of experiments on explosive mixtures of gases show that when the inflammable gas and the oxygen are mixed in the proportions which will give complete chemical combination the most explosive mixture is obtained. A mixture of such proportions is known as a *true* or *perfect explosive mixture*.

Combustion of Hydrogen and Carbon. It is a simple matter, if the chemical formula of an inflammable gas is known, to estimate the amount of oxygen necessary to give the true explosive mixture. Thus, in the case of the inflammable gas hydrogen (H_2), it can readily be shown that 1 part by weight requires 8 parts by weight of oxygen (O_2) for complete chemical combination according to the following relation—

$$O_2 + 2H_2 = 2H_2O$$

This chemical equation expresses the fact that one volume of oxygen combines with two volumes of hydrogen to form two volumes of water, or steam.

Since the atomic weights of hydrogen and oxygen are 1 and 16, respectively, it follows that the weights of two volumes of hydrogen and one volume of oxygen will be in the proportions of 1 to 8.

As we shall show, later, hydrogen (H_2) and carbon, (C) are the most important of the combustible elements,

since all the fuels used in internal combustion engines consist of these two elements, with or without the addition of oxygen. It is, therefore, of interest to consider here the combustion of carbon.

Carbon has an atomic weight of 12. It combines with oxygen according to the chemical formula—

$$C + O_2 = CO_2$$

Thus one volume of carbon combines with one volume of oxygen to form two volumes of carbon-dioxide.

Since the atomic weights of oxygen and carbon are 16 and 12, respectively, these will also be the weights of the combining gases forming the true explosive mixture. The weight of the combustion product (CO_2) will be represented by $12 + (2 \times 16) = 44$.

Similarly in the case of the explosion of hydrogen and oxygen to form water, the weight of the latter ($2H_2O$) will be $(2 \times 2) + (2 \times 16) = 36$.

It sometimes happens that there is not enough oxygen in the mixture to burn all of the carbon of the fuel, as when a petrol engine runs on a very rich (in petrol) mixture. The products of combustion then contain carbon-monoxide (CO), the molecular weight of which is 28.

TYPES OF EXPLOSION

Before proceeding with the chemical side of combustion it is necessary to point out that there are three different kinds of explosion, as follows: viz. (1) *Inflammation*, (2) *Explosion*, and (3) *Detonation*.

Inflammation. This is a relatively slow rate of burning, or explosion, the resulting flame taking a comparatively long time to spread through the mixture.

Weak mixtures of gases or vapours containing oxygen often exhibit this slow mode of burning.

In this method of combustion the burning commences from the point of ignition, and each portion of the mixture as it burns heats the neighbouring portion until its temperature is so high that it burns; and so on throughout the whole mixture.

Explosion. This is a more rapid rate of combustion of explosive mixtures. The actual rate at which the flame spreads depends upon the nature and proportions of the constituents; and upon the initial pressure at which the mixture is ignited. The greatest rates are obtained when the constituents are in about the proportions for correct chemical combination, and are ignited at high initial pressures in non-conducting closed vessels.

Detonation. This is a particularly violent method of explosion, in which the flame flashes through the explosive mixture at a very considerably higher rate than in the preceding case.

As an example of this one may mention the case of an explosive mixture consisting of equal volumes of carbon monoxide and oxygen exploded in a vessel at atmospheric pressure. If ignited by a spark a simple explosion occurs, the rate of flame travel being 3·3 ft. per second. On the other hand, if ignited, not at a single point, but at a considerable volume, detonation occurs, the velocity of flame travel being about 5,000 ft. per second.

Detonation can be made to occur in other gaseous mixtures such as hydrogen and oxygen; detonation flame rates of about 10,000 ft. per second have been measured in such cases.

Similarly, as is shown in the sections on "Fuel Technology" and "Cylinders, Cylinder Heads, and Liners," detonation may occur in internal combustion engines under certain conditions. The most frequent cause of detonation in petrol-type engines is the use

of too high a compression ratio for the particular fuel employed.

Here it may be pointed out that detonation would certainly occur in most of the modern designs of petrol engines if only pure oxygen were used. The fact that *air* is employed obviates this tendency however.

Air consists, approximately, of 4 volumes of nitrogen to 1 volume of oxygen, so that the greater proportion of the former inert, or non-inflammable, gas has the effect of modifying the nature of the explosion. The residual exhaust gases left in the cylinder at the end of the exhaust stroke also serve to dilute further the oxygen content of the incoming explosive mixture. From our present viewpoint detonation may be regarded as a wave of compression traversing the mixture, the heat of this compression igniting the mixture, so that the rate of flame travel is the same as the velocity of the compression wave in the mixture.

FLAME RATES

The three different modes of explosion having now been considered it may be of interest to mention that experiments which have been made by various investigators on gaseous mixtures ignited under different conditions established the following important facts, from the internal combustion engine viewpoint, viz. that the rate of flame travel—

1. Is increased as the temperature of the mixture is raised.
2. Is increased as the initial pressure is raised.
3. Is dependent upon the proportions of the active constituents.
4. Is diminished as the proportion of the inert gas present is increased.
5. Is much greater when the mixture is ignited at constant volume than when ignited at constant pressure.

The third item indicates the effect of mixture strength upon the rate of flame travel, and in this case, although it might at first be thought that the greatest rate of flame propagation should correspond to the mixture proportions for correct chemical combination, i.e. the

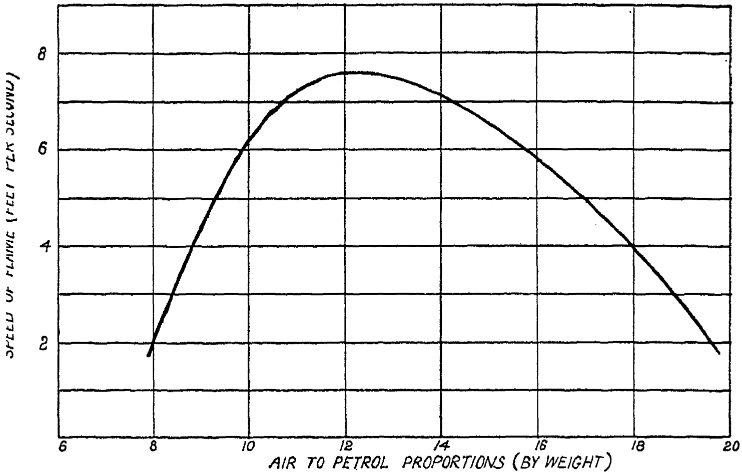


FIG. 1. SHOWING FLAME RATES FOR DIFFERENT PETROL/AIR PROPORTIONS

true explosive mixture, experimental results show that oxygen-fuel mixtures, containing a certain excess of fuel, give the greatest flame rates.

Fig. 1 shows, graphically, the relation between the flame rate and mixture strength for air-petrol vapour explosive mixtures of different proportions.

In this case the true explosive mixture, or that required for perfect chemical combustion, is one consisting of about 15 parts of air to 1 part of petrol, by weight. On the other hand, the mixture giving the maximum rate of flame travel consists of about $12\frac{1}{2}$ parts of air to 1 part of petrol, by weight. A richer

mixture than the true explosive one, therefore, gives the greatest flame rate.

This is a significant feature in petrol engine operation, for it is also known from experimental work that a mixture of air and petrol vapour of about 12 to 13 parts of air to 1 part of petrol gives the greatest mean effective pressure (or power result).

It is for these reasons that the carburettors of racing motor engines are tuned so as to give a richer mixture than that corresponding to the proportions for perfect combustion of the fuel. It will be obvious that although the greatest power is thus obtained the fuel is not used economically; in other words, the mileage per gallon will be less than if the carburettor were tuned to give the true explosive mixture.

Combustion of Hydrocarbon Fuels. As we have stated previously, most of the volatile liquid fuels used in internal combustion engines are hydrocarbons, i.e. chemical compounds of hydrogen and carbon in different proportions, represented by the formula C_xH_y.

Most petrols in present-day use consist of mixtures of hydrocarbons, namely, paraffins, naphthenes, and aromatics, having the formulae C_nH_{2n+2}, C_nH_{2n} and C_nH_{2n-6}, respectively. Examples of members of these respective series are Hexane (C_6H_{14}), Cyclohexane (C_6H_{12}), and Benzene (C_6H_6).

There is another type of liquid fuel, namely, alcohol, which contains oxygen in addition to hydrogen and carbon; thus one well-known form is Ethyl alcohol which has the formula C_2H_6O.

Having seen that the general constituents of the fuels used in internal combustion engines—whether petrol or Diesel types—are the same, let us now consider the manner in which such fuels combust when mixed with oxygen or air.

It is known from experiment that when hydrogen is

burnt, or exploded with the proper quantity of oxygen, it gives out a certain amount of heat.

Thus, when 1 lb. of hydrogen is exploded with 8 lb. of oxygen the amount of heat liberated is 62,030 British Thermal Units.*

Similarly, when 1 lb. of carbon is burnt with its proper quantity of oxygen for complete combustion, viz. $\frac{32}{12} = 2{\cdot}\dot{6}$ lb., the amount of heat evolved is 14,540 B.Th.U.'s.

It will thus be seen that the greater the proportion of hydrogen in the fuel the greater will be the amount of heat liberated when the fuel combusts with oxygen.

It is for this reason that hydrocarbon fuels which are rich in hydrogen have higher heating or calorific values.†

Estimating the Calorific Value of a Fuel. It is an instructive exercise to estimate the calorific value of any fuel, knowing its chemical formula, or composition. A typical commercial petrol has chemical formula, C_8H_{18}, this fuel belonging to the paraffin series represented by C_nH_{2n+2}.

From the atomic weights of carbon (12) and hydrogen (1), it can readily be shown that 1 lb. of the fuel contains 0·846 lb. of carbon and 0·154 lb. of hydrogen.

If these constituent weights of the fuel be multiplied by the respective calorific values given in the preceding section, the total calorific value is obtained.

$$\begin{aligned} \text{Thus calorific value of } C_8H_{18} &= (0{\cdot}846 \times 14{,}540) + (0{\cdot}154 \times 52{,}500). \\ &= 12{,}300 + 8{,}080 \\ &= 20{,}380 \text{ B.Th.U.'s per lb.} \end{aligned}$$

This is the higher heating value of the fuel.

The lower heating value allows for the latent heat of

* This value includes the latent heat of the steam formed, viz. 8,694 B.Th.U.'s. at 212° Fahr.

† Calorific values for the more important fuels are given in the section on "Fuel Technology."

the steam (1·386 lb.) which is formed. Since the latent heat per lb. is 966 B.Th.U.'s, the total latent heat must be $1{\cdot}386 \times 966 = 1{,}338$ B.Th.U.'s.

Subtracting this from the higher heating value we get

Lower heating value $= 20{,}380 - 1{,}338 = 19{,}04$. B.Th.U.'s. per lb.

It is thus a fairly simple matter to work out the calorific value of any fuel, or fuel mixture.

Combustion Calculations. It is very useful to be able to estimate the weight and volume of air required to explode, or combust, a given quantity of fuel in an engine; this calculation frequently occurs in experimental work.

We shall take the general case of a hydrocarbon fuel represented by the formula C_xH_y.

The carbon and the hydrogen are burnt to carbon-dioxide and to water, respectively, according to the relation—

$$C_xH_y + \left(x + \frac{y}{4}\right) O_2 = x\,CO + \frac{y}{2}\,H_2O \qquad (a)$$

It will be seen from this that 1 volume o: com-bines with $\left(x + \frac{y}{4}\right)$ volumes of oxygen, and after explosion yields x volumes of CO_2 and $y/2$ volumes of H_2O.

Since the composition of air, by volume, is 79·1 per cent nitrogen (N) and 20·9 per cent oxygen, it can be readily shown that 4·78 cub. ft. of air contains 1 cub. ft. of O, and 3·78 cub. ft. of N.

Hence 1 volume of C_xH_y will require $4{\cdot}78 \left(x +\right.$ volumes of air for complete combustion.

Example. What is the volume of air required to explode one volume of hexane (C_6H_{14})?

From the last formula we have $x = 6$ and $y = 14$.

Hence 1 volume of C_6H_{14} will require $4{\cdot}78\left(6 + \frac{14}{2}\right) =$ $62{\cdot}14$ volumes of air.

Next, let us consider the *weight* of air required.

The equation (a) can be expressed in molecular weights by multiplying the items therein by their atomic weights.

Thus we can obtain the molecular weights for the four expressions in equation (a) as

32; x (12 + 32) and $y/2$ (2 + 16).

It thus follows that 1 lb. of the fuel C_xH_y will require $\frac{32 + 8y}{12x}$ lb. of oxygen and will yield $\frac{44x}{12x+y}$ lb. of CO_2 and $\frac{\quad}{12x + y}$ lb. of H_2O.

Since the composition of air by weight is 76·6 per cent N, and 23·2 per cent O, it follows that 4·31 lb. of air contains 1 lb. of O and 3·31 of N.

So that 1 lb. of the fuel C_xH_y will require $4{\cdot}31 \times \frac{32x + 8y}{12x + y}$ lb. of air for complete combustion.

Example. What is the ratio of air to fuel, by weight, for the complete combustion of the fuel hexane (C_6H_{14})?

If we substitute $x = 6$ and $y = 14$ in the last expression we find that 1 lb. of C_6H_{14} will require

$$4{\cdot}31 \quad \frac{\times 6 + 8 \times}{12 \times 6 + 14} = 15{\cdot}2 \text{ lb.}$$

The ratio of air to fuel is, therefore, 15·2.

In general it will be found that most of the commercial petrols require about 15 times their weight of air for perfect combustion.

It can also readily be shown that *fuels of the aromatic*

series $C_n H_{2n-6}$, require from 13 to 14 times their weight of air for complete combustion.

On the other hand, fuels of the *naphthene series*, $C_n H_{2n}$, require about 14·7 times their weight of air.

From these considerations it follows that *as the proportion of the aromatic content of petrol is increased* less air is required for combustion.

In the above considerations we have left out any considerations of dissociation; this subject is dealt with in the section on "Fuel Technology."

Some General Notes on Combustion. Before leaving this subject of combustion it may be of some interest to mention one or two further facts which are likely to be of utility to the automobile engineer.

Volume Heating Values. It can readily be shown that in the case of petrol of the hexane class, requiring 15·2 lb. of air per lb. of fuel for combustion, the equivalent volume of air at 60° F. will be 200 cub. ft.

Further, 1 cub. ft. of mixture will give out 95 B.Th.U.'s during perfect combustion, assuming the fuel has a calorific value of 19,000 B.Th.U.'s per lb.

A convenient value to employ in petrol engine calculations is one of 100 *B.Th.U.'s per cub. ft. of mixture.*

Since 1 B.Th.U. is equivalent to 778 ft.-lb. of work, the standard mixture strength for maximum power may be written as $\frac{100 \times 778}{1728}$, or 45 *ft.-lb. per cub. in. of mixture.*

Another method of expressing the equivalent work units is $14{\cdot}8 \times 10^6$ *ft.-lb. per lb. of petrol.*

Exhaust Gas Composition. The composition of the exhaust gases from an internal combustion engine affords a reliable indication of the mixture strength or proportions.

We have already seen that if the mixture of fuel and air is such that the combustion is complete, the products

of combustion, when oxygen is used, will contain CO_2 and H_2O, or steam.

When air is used, the products will contain the inert gas nitrogen, since this takes no part in the combustion. Hence for the correct mixture strength we have in the exhaust CO_2, H_2O, and N.

If the quantity of fuel present in the explosive mixture is *less* than that required for complete combustion the mixture is said to be *weak* or *lean*. There will thus be a surplus of oxygen, some of which will pass through the engine unaffected, and will appear in the exhaust products. For a weak mixture, therefore, the products will consist of CO_2, H_2O, N, and O.

If there is *an excess of fuel* present, viz. more than required for complete combustion, some of this fuel will not be burnt completely, and will pass through to the exhaust in the form of carbon (C), or carbon-monoxide (CO).

If there is only a small excess of the fuel, the exhaust will contain CO ; if the mixture is over-rich in fuel there will also be carbon particles in the exhaust.

The rich mixture, therefore, has the following exhaust products, viz. CO_2, H_2O, N, CO, and possibly C.

There are two important facts, of interest to automobile engineers contained in these results, viz. (1) That weak mixtures are characterized by the presence of oxygen in the exhaust, and (2) Rich mixtures are indicated by the presence of carbon-monoxide in the exhaust.

It will be observed that, unless the mixture is changing in quality, or misfiring occurs, it is not possible to have both CO and O in the exhaust at the same time.

Checking Mixture Strength from the Exhaust Gases. In experimental work upon petrol and Diesel engines, it is the usual practice to collect samples of exhaust gases and to analyse these in a special gas analysis

apparatus, such as the Orsat, or Macfarlane Caldwell types. The percentages of CO_2 CO, or O in the exhaust gases are thus ascertained, and from a knowledge of the composition of the fuel the strength of mixture can be read off from a set of curves similar to that reproduced in Fig. 2.

When petrol engines are being tuned up for performance or for petrol economy, the strength of the mixture

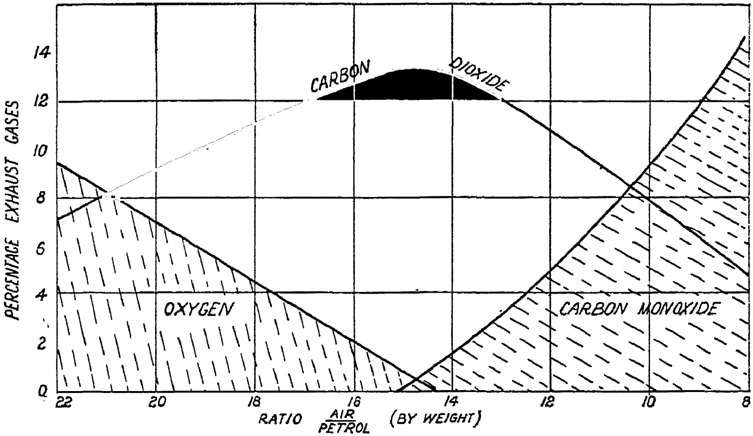


FIG. 2. EXHAUST GAS COMPOSITIONS FOR DIFFERENT MIXTURE STRENGTHS

can also be ascertained roughly by observing the colour of the exhaust flame, in a darkened room; or better still by fitting a quartz glass window in the combustion head.

If the flame is *an intense white* or bluish-white the *mixture strength is about correct* for proper combustion. If it is *distinctly blue and less intense*, the mixture is on the *weak side.*

On the other hand, if the colour of the flame is of an intense yellowish white the mixture will be on the *rich side.*

A less intense yellowish coloured flame with sooty particles (or black dust) indicates a very rich mixture.

Combustion in Closed Containers. The piston and cylinder of a petrol engine may be regarded as forming a closed vessel or container, the volume of which varies according to the movements of the piston.

At the end of the compression stroke the mixture is contained in a space representing the combustion chamber, or clearance, volume.

At this position, namely, the top dead centre of the compression stroke the piston is to all intents, stationary, so that combustion of the compressed charge takes place *at practically constant volume.*

It is for this reason that a good deal of experimental work has been carried out on mixtures of fuel vapours and air, exploded in closed vessels, in order to approach the conditions existing at the moment of ignition in the combustion chamber of an actual engine. Thus, it is possible to obtain some important information on the combustion of fuel and of gaseous explosive mixtures and of the pressures developed during and after combustion. Although, as we shall show later, the results of these closed explosion vessel tests require modification before they can be applied to actual engines, a number of interesting qualitative results have been ascertained from this type of experiment on explosion rates and pressures. It is not possible in the limited space at our disposal to describe the apparatus, or methods, of these experiments*, but only to refer briefly to the more important results.

The general method adopted in closed explosion

* For fuller information the reader is referred to the following books—

The Gas, Petrol, and Oil Engine. Sir Dugald Clerk, F.R.S. (Longmans, Green & Co.).

The Gaseous Explosions Committee (British Association) Reports.

Aircraft and Automobile Engines. A. W. Judge. (Pitman's).

vessel tests is to employ a steel bomb of spherical shape, provided with means for filling and exhausting the interior. Thermo-couples are fitted for measuring the temperatures near the inner surfaces, and special explosion pressure recording apparatus is used in order to obtain records, on a time basis, of the pressures before, during, and after the passage of the spark igniting the explosive mixture. These explosion rate

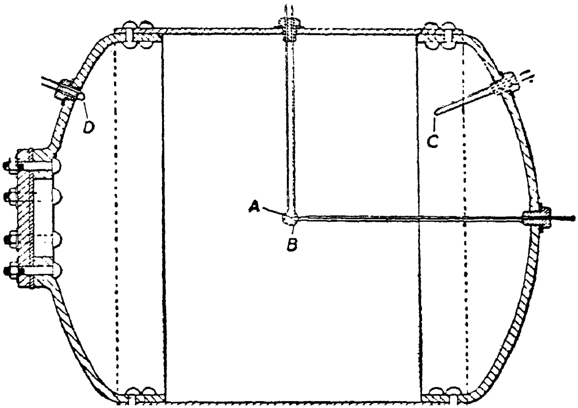


FIG 3. HOPKINSON'S EXPLOSION VESSEL.

records are of special interest in connection with the ascertaining of pressure rates for different fuel mixtures.

Fig. 3 illustrates the explosion vessel used by Prof. Hopkinson, the electrical thermometers for recording the temperatures after explosion being shown at *B*, *C*, and *D*. The electrical ignition wire is shown at *A*; this ignition point is at the centre of the vessel.

By making simultaneous readings of the thermometers in question at regular time intervals, curves showing the rates of cooling were obtained.

Explosion Vessel Results. The following are some of

the more important results obtained from closed explosion vessel experiments—

1. In closed explosion vessels the combustion is at constant volume, the portion of the mixture first ignited expanding rapidly and in so doing it projects the flame into the mass of the mixture, the remaining un-ignited portions of the mixture being compressed. The rate of inflammation at constant volume is, therefore, greater than that at constant pressure by the rate of flame projection (owing to its expansion), and also by the compression of the un-ignited portions by the part first inflamed.

2. By igniting a given quantity of mixture at several places simultaneously—by means of a long spark, or several sparks—the time of explosion can be reduced appreciably. This result is true of low-compression engines, but is hardly applicable to high compression engines of to-day, where turbulence plays a big part.

3. A comparison of the rates of cooling, after explosion, of the contents of explosion vessels of different capacities shows that the larger the vessel the slower is the rate of cooling. It is further found that if the density of the explosive mixture be increased in the same vessel the rate of cooling is diminished.

4. There is a wide variation in temperature at different parts of the explosion vessel. Thus in the case of certain experiments carried out in the explosion vessel shown in Fig. 3, the explosion pressures and temperatures were recorded by optical means, and it was found that at first the pressure rose slowly, and the temperature at the centre position, *B*, rose rapidly. The pressure rose for about 0·22 sec. at a rapid rate, and then at a decreasing rate to the maximum pressure after 0·26 sec. The temperature at *D* was found to rise suddenly after 0·23 sec.; this corresponded with the moment when the flame reached the walls of the vessel.

Further, the pressure continued to rise after the maximum temperature at the walls was attained.

The mean temperature in the vessel was 1,600° C., and at the centre *B*, the platinum of the thermometer always melted, thus showing that the temperature there always exceeded 1,750° C., the melting point of platinum. At *C* the maximum temperature was 1,700° C. whilst at *D* it was about 1,200° C.

A thermometer inserted inside, near the wall, recorded 850° C. It is evident from these results that the temperatures after the explosion of a stationary mass of explosive mixture differ considerably at different parts of the vessel.

5. In some experiments by Fenning, on fuel-air mixtures, including petrol, hexane, pentane, and benzene, at initial charge temperatures of 100° C. to 300° C. and pressures of 95 lb. per sq. in., pressure-time records were made. It was found that whilst weak and normal mixtures gave smooth explosion records, in the case of richer mixtures the maximum pressures were appreciably higher, and the explosion rates greater; in some cases detonation occurred with the richer mixture.

In regard to explosion times, petrol gave the longest and benzene the shortest period; in the latter respect the pressure records obtained from benzene were free from knock effects.

The explosion time, with different initial pressures, was found to increase with increase in initial pressure, and to decrease with increase in initial temperature. The rate of variation of explosion time with initial pressure diminished as the initial temperature was increased, being very small for air-benzene mixtures at 300° C., and also for the other fuels at 200° C.

6. *The effects of residual exhaust gases* in explosion vessels have been investigated by Grover, Fenning, and others.

Grover carried out a series of tests with explosive mixtures of coal gas and air diluted with the burnt gases of previous explosion. These tests showed that the highest pressures were obtained when the volume of fresh air admitted was only a little more than that required for complete combustion, and that if these mixture proportions were employed the charge could be diluted up to nearly 60 per cent with exhaust gases, in place of excess air.

He came to the conclusion that exhaust gases when mixed with fresh explosive mixture, in some cases, actually had the effect of increasing the maximum pressure obtained. The maximum increase of pressure due to the presence of the exhaust products was found to occur with the weakest mixture used, and the increase of pressure became less as the mixture became richer in coal gas.

In connection with these results it is possible that the exhaust gas used for diluting the fresh charge contained combustible products such as carbon-monoxide.

More recent tests made by Fleming showed that the general result obtained by adding 6·3 volumes of exhaust products to 100 volumes of air-hexane mixture was to cause an increase in the explosion time and a diminution in the explosion pressure. The percentage increase in the explosion time due to the presence of the exhaust gas was diminished as the initial temperature was increased, being about 29 per cent at 100° C. and 22 per cent at 200° C.

With air-benzene mixtures there was a greater saving in fuel consumption than with hexane due to the addition of exhaust gases.

7. In regard to *detonation*, some interesting information was brought out from Fenning's closed explosion vessel experiments. Although the conditions were different from those which occur in actual petrol engines, some useful light is thrown on the subject.

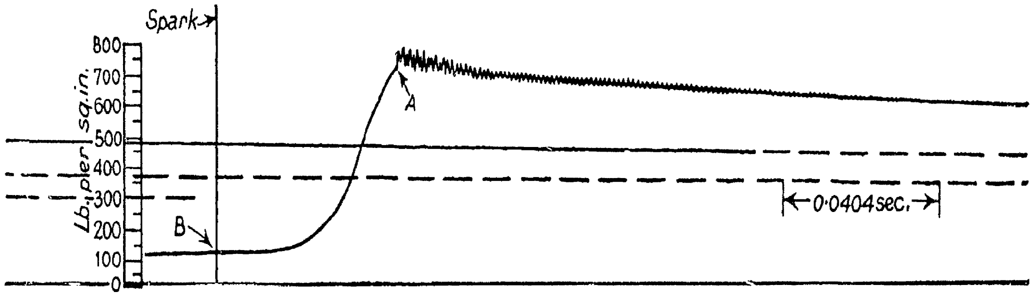


FIG 4. EXPLOSION VESSEL RECORD, SHOWING DETONATION EFFECT

Fenning's experiments indicated that in the case of a given air-petrol mixture, when the initial pressure exceeded a certain value, the pressure-time record was similar to that shown in Fig. 4.

In this case the air-petrol ratio was 12·95, the initial temperature 231° C., and pressure 128·4 lb. per sq. in. (abs.). The spark occurred at the point *B*, the pressure rising until at *A* a further sudden rise, followed by high frequency fluctuations, occurred; these conditions were found to give rise to a characteristic "knock."

It was concluded from these tests that the conditions causing the knock were apparently those pertaining to the unburnt residue at the point *A*, and not those common to the whole charge at the point of ignition.

Further, that *detonation was a temperature rather than a pressure effect*, and that any conditions that tended to impart heat to, or prevent heat loss from, this unburnt residue would tend to promote detonation. Fenning also advanced the explanation that the fact of knocking occurring more readily in large than in small cylinders with poppet-valved engines than with sleeve-valved types might possibly be explained on the ground that the cylinder temperatures are such as to result in a higher temperature of the unburnt residue in one case than in the other.

The following values of comparative detonation temperatures are given—

Pentane	486° C.
Hexane	470° C.
Heptane	446° C.
Petrol	480° C.

The Missing Pressure. It has been found from numerous closed explosion vessel experiments, carried out by recognized authorities, e.g. Clerk, Hirn, Bunsen, Mallard, and Le Chatelier that the measured values of the *maximum pressures* are only about one-half the

values calculated from a knowledge of the physical properties of the explosive mixture. The difference between the actual (or observed) pressure and the theoretical (or calculated) pressure is generally referred to as the "*missing pressure*." Similarly, if the temperature of explosion be calculated from the observed pressure rise (it cannot usually be measured by any of the direct methods used in pyrometry), it will be found that this calculated temperature is considerably lower than the theoretical temperature calculated from a knowledge of the physical properties of the mixture.

Several explanations have been put forward by different authorities in order to account for this apparent loss of energy in the exploding mixture.

It is more than probable, however, that the real explanation is that this "missing pressure" effect is due, not to any single factor, but to several of the effects, summarized briefly as follows—

(*a*) The *Increase in the Mean Specific Heat* of the products of combustion at the high temperatures existing during the explosion process.

(*b*) The *Loss of Heat to the Walls* of the explosion vessel. The latter absorbs part of the heat chiefly by radiation and to a lesser extent by conduction.

(*c*) *Dissociation Effects.* When the temperature rises above a certain value part of the carbon dioxide dissociates into carbon monoxide and oxygen, and part of the water vapour into hydrogen and oxygen. In this process of dissociation heat is absorbed from the products inside the explosion vessel, so that the maximum temperatures (and pressures) are reduced in value. It is true, however, that when the dissociated gases recombine, heat is again liberated, but this evidently does not occur until after the maximum temperature has been reached. Dissociation may here be regarded

as a suppression of heat at the maximum temperature of explosion and an evolution of heat during the expansion stroke. Its effect is to cause the actual expansion line (on the indicator diagram) to lie above the theoretical one.

(*d*) *Delayed Combustion, or "After Burning.*" This assumption suggests that combustion is not complete at the moment of maximum pressure, but continues for some part of the expansion stroke. Thus, all of the heat energy of the explosive mixture would not be developed at the time maximum pressure is attained.

Although it is possible in weak mixtures to obtain indicator diagrams of such a form as to suggest this delayed burning—and cases are not infrequent in low compression engines of combustion during the exhaust stroke—it is not considered very probable in modern high compression engines running on normal mixture strengths for after-burning to occur.

It is not regarded as possible, by some authorities, that there is any after-burning in modern petrol engines, for the spread of the flame is assisted greatly by the process of turbulence—a subject to which we shall again refer later. The shortest time interval between the passage of the spark and the occurrence of maximum pressure, in closed explosion vessel experiments, is about ·03 sec.; in actual engines, records made by indicators show that this time interval is only about one-tenth of this period.

In assessing the relative influences of the four factors we have just considered, in connection with the missing pressure, that of increased specific heat is considered the most important, and is believed to account for the greater part of the observed heat loss.

Effect of Polished Combustion Chambers. On the preceding page the effect of the loss of heat to the walls of the explosion vessel was mentioned under the

paragraph heading (*b*), and it was stated that radiation of heat from the inflamed mixture was the chief of the items of heat loss. It is, therefore, of interest to mention that the amount of heat lost to the walls depends upon the state of the surface of these walls, i.e. whether polished or coated with carbon; the latter is a good heat-absorbing medium. Hopkinson carried out some tests upon mixtures of gas and air exploded in a closed vessel, the walls of which were coated with tinfoil. The pressures developed and the rate of heat loss were then compared with those obtained in the case of the same vessel having lamp-black coated walls.

It was found that the difference in maximum pressure was very small, but that the rate of fall of pressure during cooling was considerably less with the bright lining than with the lamp-black surface.

These results are borne out by experiments made on gas engines with polished combustion chambers, a small increase in the mean effective pressure being observed.

Tests have also been made with explosion vessels coated inside with silver. In these experiments similar amounts of mixture were exploded, firstly with the interior surface coated with lamp-black, and secondly coated with silver.

It was found that in the second case the maximum pressure obtained was increased by 3 per cent, whilst the subsequent rate of cooling of the exploded products was reduced to about one-third.

It is evident, therefore, that the heat loss to the combustion chamber of a petrol engine can be reduced appreciably by polishing its interior surface; the difficulty, in practice, would be to prevent carbon deposition on such a surface.

Influence of Turbulence. Although, as we have stated previously, the results of closed explosion vessel experiments have given a good deal of valuable

information on combustion there is one very important difference between explosion vessel conditions and those which occur in petrol engines, viz. in the state of the explosive mixture at the moment of ignition. In the closed explosion vessel the mixture is to all intents and purposes at rest when it is ignited, whereas, in the case of a petrol engine the mixture is in a state of motion.

The mixture, in the latter case, is drawn into the cylinder on the suction stroke, through the inlet valve port—which is invariably placed off the centre line of the cylinder—and it enters the cylinder space at a very high velocity, viz. from 150 to 250 ft. per sec. During both the suction and the following compression stroke the mixture is therefore in a state of rapid swirling motion, or turbulence, and this state still persists at the moment of ignition. Instead, therefore, of igniting a still mass of explosive mixture from a single point, we have the case of a mass of mixture sweeping past the point of ignition at a high velocity. Evidently the flame will spread much more rapidly through the mixture in the latter case, so that one can state definitely that turbulence accelerates combustion in the case of internal combustion engines, and renders possible the very high speeds attained by modern petrol engines.

An interesting comparison of closed explosion vessel and petrol engine combustion conditions is made by Ricardo in one of his published papers read before the Aeronautical Society, in which he states that what appears to occur is substantially as follows—

"A single intensely high temperature spark passes across the electrodes (of the igniter), leaving behind a thin thread of flame. From this thin thread combustion spreads to the envelope of the mixture immediately surrounding it at a rate which depends primarily upon the temperature of the flame front itself, and to a secondary degree upon both the temperature and the density of the surrounding envelope. In this manner there grows up, gradually at first, but at a rapidly increasing rate, a

small nucleus of flame. If the contents of the explosion vessel were at rest, as in the case of an explosion vessel, this process would spread with increasing speed until it extended throughout the whole mass, but if, at any period, the rate of propagation exceeded a certain limiting figure, depending on the nature of the fuel, a detonation wave would be set up.

In the actual engine cylinder, however, the mixture is not at rest, but is being whirled about very rapidly, and is, in fact, in a highly turbulent condition. So soon, therefore, as a self-propagating nucleus of flame has been formed it is torn into fragments, which are spread and whirled about throughout the whole mass, with the result that the combustion process is speeded up enormously, as it were by handing round the fiery torch. Left to itself, the *combustion process would be far too slow to be of any use in an engine cylinder*, and though the flame temperature and rate of burning might be such as to cause detonation during the latter stages of the process, yet the process as a whole would be too slow even for an engine running at 200 r.p.m. We rely, therefore, entirely on turbulence, without which no internal combustion could run."

Apart from the important function of spreading inflammation, turbulence is also invaluable as a means of scouring away the stagnant layer of mixture which occurs on the cold cylinder walls. If left to itself this layer would either fail to combust altogether, or would burn too late to be of any practical use.

It is interesting to note that, according to Ricardo, the *wide differences in performance between the side-valve and overhead-valve types of engine* is accounted for almost entirely by the different degree of turbulence in the two cases.

In the ordinary types of engine turbulence is set up by, and depends upon the initial velocity of, the mixture through the inlet valves. Since turbulence depends upon inlet velocity it follows that it will be almost proportional to engine speed, and the time taken to spread inflammation will, therefore, also be nearly proportional, so that in terms of crank-angle the spread of inflammation will be constant.

Excessive Turbulence. It is possible by designing the shape of the combustion chamber along certain lines to obtain as much turbulence as is required; the most suitable shapes of cylinder heads are described in the section on "Cylinders, Cylinder Heads, and Liners." By following these designs further, and arranging for suitable gas velocities through the valves, it is possible to increase the turbulence considerably. Again, in the case of sleeve-valve engines, owing to the more efficient gas flow through the valve ports, it is possible to obtain all the turbulence that is required, and if one is not too careful excessive turbulence will occur.

This excessive turbulence, however obtained, is not beneficial from the point of view of engine performance, for it tends to speed up the rate of inflammation still further, to increase the heat loss to the walls and to reduce the possible range of burning. There is also a loss of power due to the additional work necessary for imparting this higher kinetic energy to the mixture. It has been shown possible, with excessive turbulence, to *prevent ignition altogether*, owing no doubt to the dissipation of the nucleus of the flame before it can be fully established at the plug points.

Experiments have shown that for each kind of fuel used in internal combustion engines there is *a definite limit to the amount of turbulence*, beyond which excessive heat loss and narrowing of the possible mixture range. The amount of turbulence which is permissible not only depends upon the fuel, but it also depends upon the initial temperature and pressure of the explosive mixture; in other words, upon the compression pressure. Thus for low compressions a higher degree of turbulence is needed than for normal compressions.

We have already mentioned that excessive turbulence causes too high a rate of pressure rise, and will now illustrate this point by referring to the results of

Ricardo's experiments on a $5\frac{1}{2}$ in. × 7 in. cylinder fitted with two different forms of combustion chamber—in one turbulence was due solely to the entering velocity of the mixture whilst in the other it was accentuated greatly, before ignition. In one case the rate of pressure rise was 20 lb. per sq. in. per degree of crank-angle; in the other nearly 100 lb. per degree.

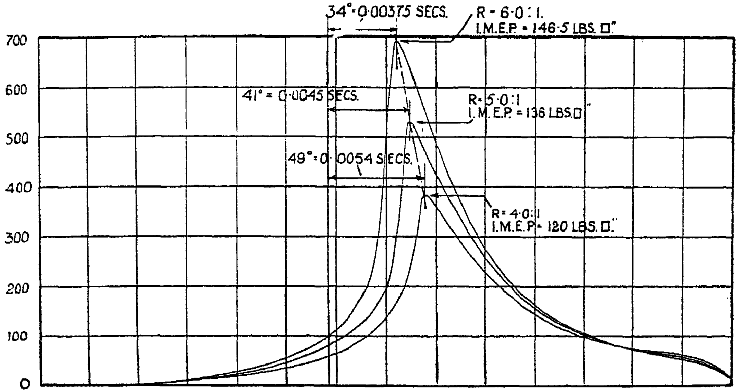


FIG. 5. THREE INDICATOR DIAGRAMS ON A TIME BASE

Showing the time occupied from point of ignition to maximum pressure, in degrees of crank-angle and in seconds. These were taken at constant mixture strength and with ignition 30° before top centre in each case. The speed was 1,500 r.p.m.

The three diagrams shown in Fig. 5 were taken from a Ricardo variable compression engine with fixed ignition, constant mixture strength and constant speed, but at the compression ratios of 4 : 1, 5 : 1, and 6 : 1. Here, in all cases, the turbulence must be practically similar, but there is evidently a considerable speeding up of the whole process as the compression is raised, for the time-intervals between the passages of the sparks and of maximum pressure are, respectively, ·0054, ·0045, and ·00375 seconds.

In concluding these considerations on turbulence it should be mentioned that the phenomenon in question was investigated a long time ago, in the case of gas-engines, by Hopkinson and Dugald Clerk.

Direct Combustion Observations. In the past a good deal of our theories of the combustion process in petrol engines was of a more or less conjectural nature, due to the extreme difficulty of making any direct observations and measurements of the combustion process under practical working conditions.

A few years before the Great War the writer, in collaboration with the late Prof. W. Watson, F.R.S., obtained some interesting spectrographic records of the combustion process, by means of a spectroscopic camera, stroboscopic disc, and a quartz window fitted to the cylinder head. It was possible to obtain spectroscopic photographs at any part of the piston's stroke before and after ignition, and to observe the colours and intensities of the flame in the combustion head. Since then other investigators, in Germany and the U.S.A., have fitted several quartz windows to cylinder heads, and have examined the flame characteristics, using a stroboscopic disc; i.e. a disc with a viewing slit, driven at engine speed through a differential gearing—in order to alter its phase relationship to the engine.

Motion picture films have been taken recently to show the spread of the flame from the sparking plug to the extremities of the cylinder head. This method has been used to demonstrate the less violent explosion in the case of doped fuels, e.g. tetra-ethyl lead, than in ordinary fuels used under similar conditions of compression and speed.

Thermal Efficiency. It will be evident, from previous statements, that when a given quantity of fuel is mixed with its correct proportions of air and ignited under compression in a petrol engine cylinder, that only a part

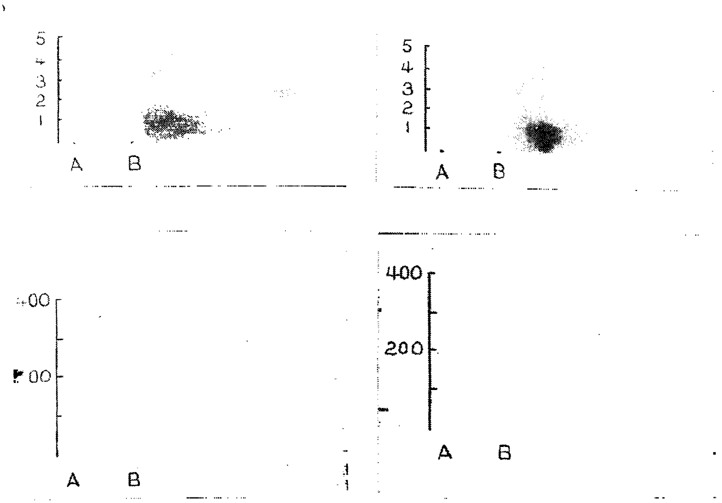
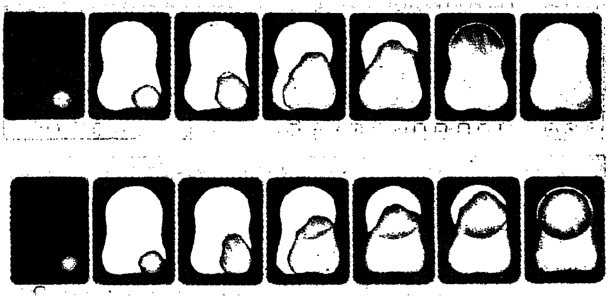


FIG. 6. SIMULTANEOUS FLAME AND PRESSURE PHOTOGRAPHIC RECORDS OF AN ACTUAL EXPLOSION IN A PETROL ENGINE CYLINDER

The two upper illustrations are flame photographs made through a quartz window extending across the top of combustion chamber. The film moved from right to left, and flame from bottom to top of picture. Lower illustrations are the corresponding pressure records. *A–B* is the compression stroke, *B* being the point of ignition. The left illustrations are for ethyl petrol, and show the smoother combustion effects as compared with those of ordinary petrol (on right).



B

FIG. 7. CINEMA FILMS, TAKEN THROUGH A QUARTZ WINDOW IN CYLINDER HEAD

Showing the spread of the flame from the moment of ignition as shown on the extreme left. *A* corresponds to ordinary petrol, whilst *B* refers to ethyl petrol, which gives more regular pressure rise and combustion spread.

of the heat contents of the fuel are utilized in performing useful work upon the piston, due to various losses of heat; these losses are dealt with more fully in the section on "Fuel Technology."

In assigning the performance of any type of prime mover, or engine deriving its source of power output from any fuel, it is usual to express the actual amount of useful work obtained from a given quantity of fuel in terms of the amount of work equivalent to the total heat of this same quantity of fuel. This ratio is termed the Thermal Efficiency, or T.E.

$$\text{Thus thermal efficiency} = \frac{\text{Useful work obtained}}{\text{Heat of fuel supplied}}$$

Both of these quantities must be expressed in the same units, i.e. work, or heat units.

The useful work in question is that developed in the cylinder. This is the "work" obtained from measurements of indicator diagrams. It is termed the *Indicated Work*, and the thermal efficiency is known in this case as the *Indicated Thermal Efficiency.*

If the useful work is considered as that obtained at the crankshaft, or from brake horse-power tests, the thermal efficiency is then known as the *Brake Thermal Efficiency*; its value is obviously less than the former efficiency.

The ratio of the brake to the indicated T.E. is the same as the ratio $\frac{\text{b.h.p.}}{\text{i.h.p.}}$ that is, the *Mechanical Efficiency.*

Calculating the Thermal Efficiency. If the fuel consumption and indicated (or brake) h.p. be known from test results the T.E. can readily be calculated. Thus, denoting the indicated horse-power by i.h.p., the fuel consumption of W lb. per hour, and the calorific value of the fuel by C British thermal units per pound, we have—

$$\text{Indicated T.E.} = \frac{\text{i.h.p.} \times 33{,}000 \times 60}{W \times C \times 778}$$

Here we have converted the heat units in the lower expression to work units, since 1 B.Th.U. is equivalent to 778 foot-pounds.

This expression reduces to the following—

$$\text{Indicated T.E.} = 2540 \, . \, \frac{\text{i.h.p.}}{WC}$$

If the mechanical efficiency be denoted by the usual symbol η we have—

$$\text{Brake T.E.} = \frac{2540\eta \, . \, \text{i.h.p.}}{WC}$$

Example. What is the brake thermal efficiency of a motor-car engine having a fuel consumption of 0·60 lb. per b.h.p. hour, the fuel having a calorific value of 18,000 B.Th.U.'s per lb.?

$$\text{Brake T.E.} = \frac{2540 \times 1}{\times 18{,}000} = \frac{1270}{5400} = \cdot 235$$

or 23·5 *per cent.*

If the mechanical efficiency of the engine in question were 85 per cent, the indicated T.E. would be

$$\frac{23 \cdot 5}{\cdot 85} = 27 \cdot 6 \textit{ per cent.}$$

Notes Upon Thermal Efficiencies. The above expressions for thermal efficiencies are true for all types of engines, or prime movers, deriving their power from fuels; these include gas, steam, oil, and petrol engines. A large amount of experimental work has been carried out in connection with petrol engines, with the result that our knowledge of the factors which influence its performance is now fairly complete. We are, therefore, able briefly to enumerate here some of the more

important factors which influence the thermal efficiency of a petrol-type engine.

THE COMPRESSION RATIO. It has already been shown in the section on "Thermodynamics" that the air standard efficiency of an engine working upon the Otto cycle increases as the compression ratio is increased, or, in other words, for a given heat input more useful work is obtained from the engine.

In a similar manner the effect of raising the compression ratio of a petrol-type engine is to raise the thermal efficiency, for more useful work is obtained from a given fuel consumption. Increasing the compression within the detonation limits of the fuel used reduces the heat losses, since a smaller relative area of combustion chamber is exposed to such losses. Moreover, the pressure rise period is reduced, so that the time of this part of the heat loss is also reduced.

The expansion is also greater, so that more useful work is obtained from the hot products of combustion.

The manner in which the observed thermal efficiency varies with the compression ratio is shown in Fig. 5 in the previous section, entitled "Fuel Technology." There the efficiency increases from about 27·5 for a compression ratio of 4 : 1 to 37·5 for one of 7 : 1.

NATURE OF THE FUEL. Each type of fuel used in an engine working on the Otto cycle under the same compression, speed, and mixture strength conditions, has its own particular value of maximum thermal efficiency. Thus benzol gives a rather higher value than petrol, and alcohol than benzol.

If, however, each fuel is used at its *highest useful compression ratio*, or H.U.C.R., the differences become more marked, since much higher compressions can be used with alcohol than with benzol or petrol, whilst benzol has a higher H.U.C.R. than petrol.

For example, alcohol will work at compression ratios

of at least 8 : 1, whereas aromatic-free petrol will not operate above 4·85 : 1. The best "aromatic" petrol has an H.U.C.R. of about 6 :1, whilst benzol is between 6 : 1 and 7 : 1. The increase of thermal efficiency due to the compression effect alone is, therefore, appreciable for alcohol and benzol. One example of this may be quoted, viz. in the case of some tests made by Ricardo on a single cylinder engine with 95 per cent alcohol and with ordinary petrol as fuels. In the former case the indicated T.E. was 37·5 per cent, and in the latter 27·5 per cent, under the same conditions.

MIXTURE STRENGTH. The thermal efficiency for any given fuel depends upon the proportions of air to fuel. The greatest efficiencies are obtained from mixtures about 15 per cent weaker than those giving perfect combustion of the fuel. Further, the richer mixtures giving the maximum power results have lower thermal efficiencies than that of the correct mixture. Stated in another way, the fuel consumption of engines working on richer mixtures will be higher per horse-power developed than for weaker ones.

THE ENGINE LOAD. As a petrol engine is throttled down, without altering the ignition setting, the efficiency falls progressively, but if at each throttle opening *the ignition timing is advanced correspondingly*, the thermal efficiency remains practically constant.

Some tests made by Ricardo, using both alcohol and petrol as fuels, showed that in the latter case the thermal efficiencies were practically constant for loads ranging from 30 per cent to full load (100 per cent).

SIZE OF CYLINDER. It was shown by Callendar, from theoretical considerations that the relative efficiency of different engines could be expressed in the following manner—

$$E_r = E\left(1 - \frac{A}{D} - \frac{B}{nD}\right)$$

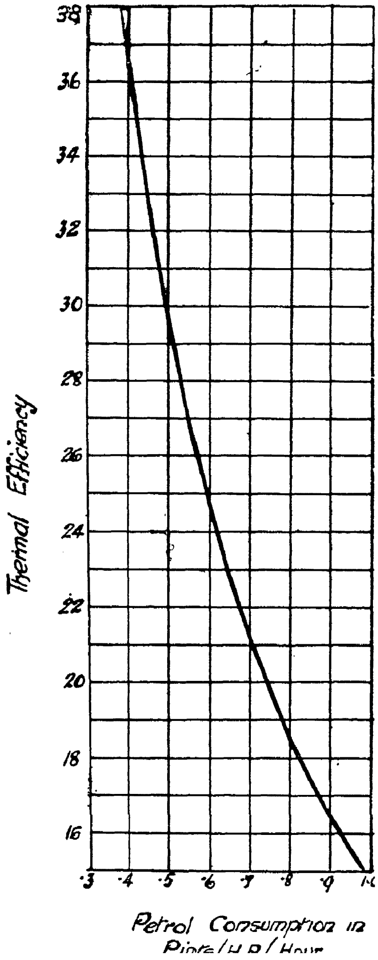


Fig. 8. Graph Showing the Relation Between the Thermal Efficiency and Petrol Consumption (Ricardo)

where E is the theoretical limit of attainable efficiency in practice, and D the diameter of the cylinder. A, B, and n are constants.

The general deduction one can make from this formula is that the thermal efficiency of a petrol engine increases as the diameter of its cylinder is increased for similar designs of engines working at the same piston speeds.

Other factors which affect the thermal efficiency include the *Design of the Combustion Chamber, The Valve Timing, The Engine Speed, Ratio of Bore to Stroke,* and *Material of Cylinder.* Space considerations will not permit any detailed discussion of these factors, but it will be obvious that in making comparisons of different engines, each of these factors should be taken into account, or the conditions selected so that comparisons are made under similar circumstances.

Thermal Efficiency and Fuel Consumption. Automobile engineers will find the graphical results given in Fig. 8 useful as a means of ascertaining rapidly the equivalent fuel consumption, expressed in pints of petrol per b.h.p. hour for any given brake thermal efficiency value. Alternatively if the fuel consumption is measured the corresponding thermal efficiency can at once be found from the graph.

This curve is based upon a calorific value of 18,290 B.Th.U.'s per lb. of petrol. One pint of petrol per hour is equivalent to 6·76 b.h.p., and one pint per hour gives 14·8 brake T.E.

Thermal Efficiency and Air Consumption. It has been shown by Ricardo, from the results of a number of engine tests, in which air and fuel consumption measurements were made, that there is a definite relationship between the indicated M.E.P. and the weight of air entering the cylinder. This relationship has been found to hold good over the range of mixtures of from 5 to 35 per cent rich in fuel.

The thermal efficiency can be determined with a fair degree of accuracy in practice by adjusting the carburettor so as to give a mixture for maximum power. The mixture in this case will be about 20 per cent rich in fuel.

The efficiency can then be found from the following relation—

$$\text{Indicated thermal efficiency} = \frac{\text{I.H.P.}}{W_x \times C}$$

where W_x = lb. of air consumed per hour, and C is constant depending upon the kind of fuel used. Its values for different fuels are given on page 138.

It was also shown by Ricardo that no matter what type of fuel was used the following relation between the air consumed and the heat liberated during combustion always held.

Heat liberated per lb. of air = 1,300 B.Th.U.'s.

Rich Mixtures of	Value of Constant C
Petrol	1·96
Hexane	1·94
Heptane	1·955
Ether	1·975
99% Ethyl Alcohol	2·02

Comparative Thermal Efficiencies. In connection with the foregoing considerations relating to the thermal efficiency of a petrol engine, it is interesting to compare the performance of this with other types of engine deriving their ultimate power output from the combustion of a fuel.

We may legitimately include the steam engine under this heading, for the original source of its heat supply is that of the combustion of coal in its boiler.

Hitherto, the steam engine has had a very poor performance as compared with the internal combustion engine, but in recent times by the use of high steam pressures, high degrees of superheat, steam turbines working with low discharge pressures, and by using special heat recuperating plant it has been shown possible to attain the same, if not better, thermal efficiencies as in the case of petrol engines.

The usual range of brake thermal efficiency for the latter engines is 23 to 28 per cent. In the case of recent marine steam engines using steam at pressures of 1,800 lb. per sq. in., and steam temperatures of 700° F. to 1,000° F., thermal efficiencies of 27 to 35 per cent are attainable.

The Diesel type of engine, on account of its higher compression ratios, viz. from 12 : 1 to 16 : 1, shows a marked superiority in efficiency over the petrol engine,

for the usual range of brake thermal efficiencies is from 32 to 38 per cent. It is here of interest to note that apart from obtaining a greater part of the heat energy of the fuel as useful power, the Diesel engine operates on low-grade fuels, which are, to-day, much cheaper than low flash point fuels such as petrols.

A still more efficient engine developed from the Diesel engine is that known as the Still engine. This type utilizes most of the waste heat of the exhaust gases to raise steam in a separate boiler. This steam is led to the opposite side of the piston to the Diesel engine side, so that it does useful work on the piston during the exhaust and compression strokes of the (four-cycle) Diesel engine side. The brake thermal efficiency of this type of engine is 38 to 44 per cent.

Maximum Compression Ratios in Practice. The general tendency in the development of petrol engines is to increase the power output per unit cylinder capacity, in order that the engine may be as light as possible and, at the same time, of minimum bulk for its power output.

The use of special metals in engine construction, such as magnesium and aluminium alloys and heat-treated alloy steels and irons, has enabled the designer to reduce the weight of the engine considerably so that, to-day, the engine weight for a given compression ratio represents almost the limit compatible with considerations of safety, reliability, and satisfactory wearing qualities. Very little further progress can therefore be anticipated in so far as engine materials, at present available, are concerned. The main avenue of progress still open to the engine designer is that of obtaining increased performance from an engine of given cylinder dimensions by an increase in the compression ratio, so that not only the mean effective pressure (or power output) but the thermal efficiency of the engine

will be increased, in the case of unsupercharged engines. In the latter connection, it is, of course, possible to increase the power output per unit cylinder capacity by compressing the fuel-air charge before admission to the cylinder; in this manner the maximum power can be increased, for a given size of engine, up to at least 50 per cent.

The difficulties hitherto encountered in regard to the use of higher compression ratios have been of a practical nature, for it was discovered at an early stage that if the compression ratio was gradually increased in the same engine, a stage was reached at which serious detonation effects occurred; the maximum compression that could therefore be employed must always be lower than that at which detonation occurred.

The use of special fuels and fuel mixtures, however, has enabled the detonation effects to be overcome to such an extent that much higher compression ratios are now possible than hitherto. Thus, with modern commercial petrols, compression ratios of 5·5 to 7·0 to 1 are now employed for motor car engines, whereas before, ratios were limited to about 5·0 to 5·5 to 1.

It has been shown that the higher the *octane rating*[1] of the fuel the less is its detonation tendency, so that the higher will be the value of the maximum compression ratio that can be used in practice. Further, it can be stated that the greatest advances which have been made in connection with petrol engine performances from given sizes of cylinders have been due to the raising, progressively, of the octane ratings of the fuels used.

Aluminium Cylinder Heads. Some notable progress, but of a lesser extent, has also been made in connection with the improved performance due to the use of

[1] See p. 148.

special shapes of combustion chambers, giving higher thermal efficiencies; also with the adoption of aluminium alloy cylinder heads. The latter enable higher compressions than are possible with cast-iron heads to be employed, on account of their better heat conductivity.

It may here be of some interest to quote the results of tests made by the research department of the Aluminium Society of America upon *a side-valve engine fitted with cast-iron and aluminium alloy cylinder heads.* The maximum value of the compression ratio that could be employed for the aluminium head was 6·5 : 1, and for the cast-iron one, 5·0 : 1. The maximum brake horse-powers developed in the two cases were 92 and 80·5 and brake M.E.P. values, 98 and 81 lb. per sq. in., respectively.

Further tests made upon the same engine, but fitted with a combustion chamber of special design, showed that it was possible to obtain a maximum brake M.E.P. of 125 lb. per sq. in. with the aluminium cylinder head and 15 per cent more power than with the best cast-iron head, with no greater roughness or tendency to detonate than with the cast-iron one; the maximum power in each case was developed at a speed of 3,000 r.p.m.

Efficiency and Compression Ratio. It is of interest to consider the possible gains in thermal efficiency as the compression ratio is increased progressively in order to define the possible limiting ratio beyond which there is little to be gained, from the efficiency point of view, although the power output may not be limited. If, for instance, the compression ratio is increased from 4·0 : 1 to 5·0 : 1, the theoretical thermal efficiency will be increased by about 11·5 per cent. A further raising of the ratio from 5 : 1 to 6 : 1 will give an increase in the efficiency of about 7·8 per cent, whilst

the increase in efficiency from 6 : 1 to 7 : 1 is 5·5 per cent and from 7 : 1 to 8 : 1 only 4·6 per cent.

From these results it will be seen that the percentage gains in efficiency become progressively smaller as the compression ratios are increased. From the point of view of the maximum explosion pressures developed by a given mixture strength under similar conditions, but for different compression ratios, the increases are considerably greater as the compression is raised. Thus, for compression ratios of 4, 5, 6, 7, and 8 to 1, the corresponding maximum explosion pressures for petrol-air mixtures in a certain engine were 360, 490, 625, 770, and 930 lb. per sq. in.

It is significant from these results that when the compression ratio is raised from 4 : 1 to 5 : 1, the efficiency is increased by 11·5 per cent and the maximum pressure increased by 130 lb. per sq. in. When the compression is raised from 7 : 1 to 8 : 1, the efficiency is increased by only 4·6 per cent but the maximum pressure goes up by 160 lb. per sq. in.

Since the maximum pressure defines the size, or weight, of the stress-bearing members of an engine, it will be evident that the relatively small gains in efficiency at the higher compression values mentioned must be obtained at the expense of an appreciable increase in engine weight.

The designer must therefore make a compromise when selecting the compression ratio so that the engine weight will not be excessive; the employment of relatively high speeds with medium compression ratios probably give the most satisfactory performances for a given size and weight of engine.

The limiting compression ratios, apart from other considerations, are also governed by the diameter of the engine cylinder, and to some extent by the stroke-bore ratio. It was demonstrated by the late

Prof. H. Callender and other experimenters that the heat losses are relatively greater in smaller than in larger engines, so that *a priori* higher compression ratios can be employed in small than large engines. Ricardo, who has studied this problem, gives the following values for the limiting compression ratios using one specific type of petrol, for commercial type car engines—

Compression Ratio . .	6·5	7·0	7·5	8·0
Cylinder Bore (inches) .	4 to $5\frac{1}{2}$	3 to 4	$2\frac{1}{2}$ to 3	2 to $2\frac{1}{2}$

These values do not necessarily mean that engines having cylinders of the dimensions given will not operate upon higher compressions, but that if 6·5 : 1 is the limiting ratio for the larger size mentioned, the higher values will be applicable to the other sizes given.

It is possible, of course, by using special fuels of the "doped" kind, to employ compression ratios up to 12 : 1 in specially designed engines.

Volumetric Efficiency. In all practical considerations of combustion in petrol engines, and in connection also with the calculation of cylinder temperatures, pressures and power outputs, it is necessary to take into account the fact that the cylinder is never completely filled with fresh mixture at the end of the suction stroke. Owing to various causes, which we shall consider later, the actual weight of charge drawn into the cylinder is always less than the theoretical quantity required to fill it.

By employing devices known as superchargers, however, the initial charge can be delivered *under pressure*, instead of under the suction effect of the piston on its

inlet stroke; we shall not, however, consider this aspect of the subject in the present considerations.

The term "Volumetric Efficiency" is usually defined as the ratio of the actual weight of fresh mixture, or charge, at atmospheric pressure and temperature, to that which would completely fill the working volume.

It can readily be shown that *the density of the charge* in the cylinder at the end of the suction stroke is a direct measure of its volumetric efficiency; it is, therefore, only necessary to compare the density of the charge with the density of the air outside the engine in order to obtain the volumetric efficiency. The following are, briefly, the factors influencing volumetric efficiency in the case of high speed petrol engines

THE DESIGN OF THE CARBURETTOR AND INDUCTION PIPE SYSTEM. If these have long and tortuous bends, the efficiency will be low, for the charge moves through these passages at a very high velocity, viz. from 100 to 150 m.p.h., at high engine speeds. Further, if the inside surfaces of the passages are *rough*, there will be a loss of efficiency, due to increased surface friction.

THE VALVE DESIGN AND VALVE TIMING. For the maximum volumetric efficiency the valve should be of ample diameter and lift, so that the full amount of charge can pass through the opening without wire-drawing. In special cases the valves are each duplicated.

Further, at the maximum designed speed of the engine, the inlet valve should have as *long a period of opening* as possible. In this respect, for normal motor car engines the inlet valve usually opens on top dead centre, and closes about 30° to 40° past bottom dead centre.

In the case of racing engines, however, at the higher speeds it is necessary to arrange for longer periods of opening, expressed in terms of crank-angle. Thus the

inlet valve usually opens at 10° to 15° before top centre, and closes 50° to 60° past bottom dead centre. Overlap of the opening of the inlet and closing of the exhaust valve is also arranged, so that the inertia effects of the exhaust products and of the fresh charge can be utilized beneficially.

THE HEATING OF THE CHARGE. If the incoming charge is heated appreciably during its passage into the cylinder, its density will be reduced, and its volumetric efficiency lowered. The presence of hot exhaust gases tends to heat the charge, so that adequate scavenging of the gases is essential to high volumetric efficiency.

Gas Velocity. The velocity of the fresh charge through the inlet passages is of great importance in petrol engine design, for it is the maximum permissible value of this velocity that fixes the design and dimensions of the valves and valve ports.

It is possible to calculate the gas velocity if we know the pressure, at the end of the induction stroke, inside the cylinder, from the relation—

$$\text{Velocity} = \sqrt{\frac{2g}{\rho}(P - P_o)}$$

where P = atmospheric pressure, and P_o the pressure inside cylinder, in lb. per sq. in., g = acceleration due to gravity and ρ = density of the charge. The value of P_o can be obtained from the indicator diagram.

EXAMPLE. Calculate the gas velocity, assuming the pressure of the charge at the end of the induction stroke is 13·7 lb. per sq. in., in the case of an air-petrol mixture of 15 : 1 proportions.

Taking atmospheric pressure as 14·7 lb. per sq. in., and the density of the mixture as ·090 lb. per cub. ft., we have—

$P = 14{\cdot}7 \times 144$ lb. per sq. ft.
$P_o = 13{\cdot}7 \times 144$,, ,, ,, ,,

$g = 32{\cdot}2$ ft. per sec. per sec.
$\rho = {\cdot}090$ lb. per cub. ft.

Then $v = \sqrt{\dfrac{64{\cdot}4\,(14{\cdot}7 - 13{\cdot}7) \times 144}{{\cdot}09}} = 320$ ft. per sec., approximately.

In practice, the inlet velocity varies from about 120 to 240 ft. per sec., the latter value applying to high speed engines of the racing class. In order to keep this velocity down as far as possible, it is sometimes the practice to employ two inlet valves per cylinder; in such cases the exhaust valves are also duplicated.

A useful graph for the automobile engine designer is that shown in Fig. 9. The curves show the relationship between the volumetric efficiency and the gas velocity for short induction pipes.

The curves are based upon certain minimum heating losses. Thus, in the case of the gas velocity, 100 ft. per sec., the volumetric efficiency of 80·5 per cent shows a loss of 19·5 per cent made up as follows—

Heating of mixture at inlet valve		17·3 per cent.
Pipe resistance . . .		2·2 ,, ,,
	Total loss	19·5 ,, ,,

The dotted horizontal lines refer to compression ratios r denoted by the figures given.

The curve A has been ascertained for engines with the valves in the cylinder head, i.e. the overhead valve type. The latter usually shows a higher volumetric efficiency than the side-valve type to which curve B refers.

The results shown by the curves in Fig. 9 represent the best values obtainable, or possible, in practice, with engines of the non-supercharged type working on the four stroke principle. It will be observed that even

under the best conditions the volumetric efficiency is never more than 80 per cent.

Maximum Power and Oxygen Content. An important

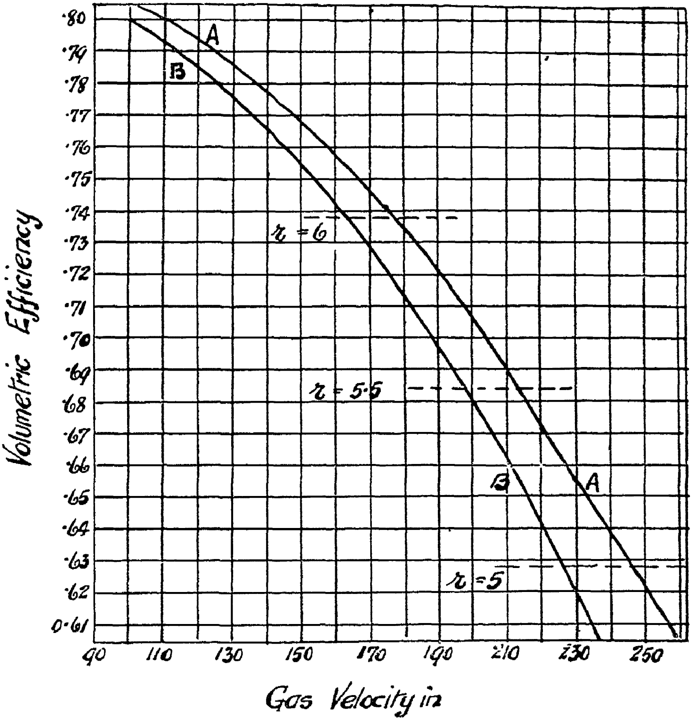


FIG. 9. SHOWING RELATION BETWEEN VOLUMETRIC EFFICIENCY AND GAS VELOCITY

factor in considering the possible performances of new types of internal combustion engine is that of the maximum amount of power which is possible from a given size of cylinder or piston displacement.

The amount of combustion energy obtainable from a

given quantity of fuel-air mixture will depend, primarily, upon the total amount of oxygen that is present, for this quantity fixes, definitely, the amount of fuel which can be combusted.

If V_s is the volume of the piston displacement in cub. ft., and v that of the combustion head,

Then, volume of fresh charge induced

$= E_v \,.\, V_s$ cub. ft., where $E_v =$ volumetric efficiency.

The amount of oxygen present is

$$= 0{\cdot}232 \times {\cdot}08073\, E_v \,.\, V_s \text{ lb.} = {\cdot}01873\, E_v \,.\, V_s \text{ lb.}$$

And the weight of petrol for complete combustion

$$= \frac{{\cdot}01873\, E_v \,.\, V_s}{3{\cdot}53} \text{ lb.} = {\cdot}005305\, E_v \,.\, V_s \text{ lb.}$$

The maximum available heat energy in this amount

$$= 18{,}700 \times {\cdot}005305\, E_v \,.\, V_s \text{ B.T.U.'s}$$
$$= 99{\cdot}19\, E_v \,.\, V_s \text{ B.T.U.'s.}$$

We are thus led to the conclusion that under ideal conditions, when the volumetric efficiency is unity ($E_v = 1$), the ideal maximum available heat energy for the petrol-air mixture giving complete combustion is 99·19 *B.Th.U.'s per cub. ft.* If this quantity is multiplied by the thermal efficiency, the *actual* number of heat units liberated as useful work is obtained.

Octane Value of Fuels. The tendency or otherwise of a fuel to detonate is now denoted by what is known as its *Octane Value*. In this method the behaviour of a standard engine operating upon the given fuel is compared with that of the same engine running upon a fuel mixture of two specially selected fuels, namely, *iso-octane* and *heptane*, which have definite and constant physical properties.

Iso-octane is a fuel which has a higher anti-knock value than that of all commercial grades of petrol,

whilst heptane is a poor fuel from the point of view of anti-knock performance and can only be used in low compression engines.

It is therefore customary to consider iso-octane as the ideal standard for high anti-knock properties and to give it the arbitrary value of 100, whilst heptane is defined by a value of 0.

All other fuels are compared with mixtures of iso-octane and heptane having the same anti-knock properties in the test engine.

Thus if it is found that a mixture consisting of 78 parts of iso-octane and 22 parts of heptane gives the same anti-knock properties as those of the fuel to be assessed, when tested under identical conditions in the standard detonation test engine, this fuel is said to have an octane value, or number, of 78.

In practice the relatively expensive iso-octane and heptane fuels are often replaced by other sub-standard petrols and mixtures of these petrols with certain anti-knock constituents, such as benzene and tetra-ethyl lead, the octane values of this range of fuels being determined by comparison with the standard mixtures of iso-octane and heptane.

It is now customary in aircraft fuel specifications to stipulate the octane values of the fuels. Thus the Air Ministry's D.T.D. 143 fuel has an octane value of 75·5 to 76·0, whilst the D.T.D. 230 fuel value is 87.

Later fuel specifications issued in this country and the U.S.A. provide for octane values of 90 and above; thus the D.T.D. 134 and 5 has an octane value of 90. The higher octane value fuels contain tetra-ethyl lead, and in this respect the percentage of this "dope" is limited to about 4 c.c. per gallon of fuel, since higher proportions tend to introduce problems not encountered in the case of ordinary fuels, including corrosion of the sparking plugs, valves, valve seatings, and cylinders.

It may be mentioned that the anti-knock properties of standard comparison fuels are determined on a standard engine known as the Co-operative Fuel Research or C.F.R. engine, which is provided with an apparatus known as a "knockmeter" of the bouncing pin type for determining the knock intensity.

Liquid Cooled Engines. In recent years a petrol engine cooling system, known as the liquid cooled engine one, has been introduced with satisfactory results. Instead of using ordinary water for cooling purposes, its place has been taken by a liquid having a higher boiling point, known as ethylene-glycol. This liquid has a boiling point of about 197° C. for the "Prestone" grade.

With ethylene-glycol the compression ratio can be increased, and engines can be made lighter than water-cooled and also air-cooled types. Another important advantage lies in the much smaller radiators that can be used, namely, about one-third the size of the water-cooled type.

The method in question thus offers special advantages in connection with high performance aircraft engines and has therefore been adopted in many recent military aeroplanes.

SECTION IV

CYLINDERS, CYLINDER HEADS, AND LINERS

BY

S. W. NIXON, M.Sc., A.M.I.A.E.

SECTION IV

CYLINDERS, CYLINDER HEADS, AND LINERS

The Cylinder Block. Modern automobile engineering calls for highly complicated castings, the material specifications of which are almost wholly governed by foundry requirements. This is particularly the case as far as the cylinder block is concerned. With very few exceptions cast iron is the universal material for block castings. In the modern engine without cylinder liners the block combines the function of an important part of the engine structure, with the provision of bearing surfaces for the pistons. The requirements with regard to material in order satisfactorily to perform these functions are that the material shall—

1. Cast readily, giving a homogeneous casting free from blow-holes.
2. Permit of rapid and accurate machining.
3. Be as hard as possible, taking into account items 1 and 2.

Variations in the surface hardness of block castings are largely due to variations in the section thickness and to the rate of cooling. This results in a tendency for the hardness to be a minimum in the bore, where it should be at a maximum value compatible with satisfactory machining properties of the casting in general. On this account a minimum hardness is frequently specified for the cylinder bore when purchasing cylinder blocks. Increase in hardness of the bore is usually accompanied by increased difficulty in machining the outside of the casting. This condition can be met by the use of suitable alloy cast irons.

The slightest porosity has to be avoided in cylinder block manufacture. Unfortunately, blow-holes and porous places may not be evident until after several machining operations have been performed. On this score aluminium is a better foundry material than cast iron, but this advantage is offset by the higher cost of the former material. On this account, therefore, aluminium is but rarely specified for automobile engine cylinder blocks.

The difficulties met with in cylinder block manufacture have been intensified of recent years by constructional developments, and by the demand for longer life from the bore, from the point of view of wear. Constructional developments include the almost universal adoption of monobloc construction, in some cases extending to V type twelve and sixteen cylinder engines; the combination of the cylinder block and crankcase in a single unit, cast-in manifolds, and complicated valve port arrangements in six and eight cylinder engines.

The demand for better wearing properties of the cylinder bore has arisen on account of the increased use of aluminium pistons, higher piston speeds, greatly increased mileages, particularly in commercial vehicle work, and the demand for sustained low oil consumption.

In private car and passenger vehicle engines the practice of casting a number of cylinders in one block has been almost universally adopted. While certain types of commercial vehicle engines still have two or three cylinders per block, in general this development has proceeded to the wholly monobloc cylinder casting. The advantages of monobloc construction include the following—

1. It lends itself to modern production, thereby greatly reducing the cost. This applies in particular to

machining operations, such as milling or grinding the main cylinder head and crankcase joint faces, manifold joint faces, cylinder boring, and flange drilling, etc.

2. It facilitates the use of simple manifolding systems.

3. The number of joints is greatly reduced, particularly in the water system.

4. It allows more positive water circulation.

5. It provides freedom for designs not otherwise possible, e.g. overhead or high camshaft location.

6. It permits closer spacing of cylinders.

7. It provides a much more rigid assembly, especially when combined with a deep crankcase.

For small engines, particularly with four cylinders, where low cost, clean design, and freedom from leakage are important factors, the adoption of induction manifolds cast in the block or cylinder head is advantageous. It is accompanied, however, by certain technical disadvantages. Freedom in manifold design is sacrificed, since it is limited by foundry requirements. The heat supplied to the mixture in its passage to the cylinders is supplied from a source of relatively low temperature, and merely reduces the volumetric efficiency without causing any real vaporization of the heavier fractions of the fuel. In the case of modern six and eight cylinder engines, the demand for satisfactory operation from cold calls for an effective hot spot, and external manifolding is incorporated to meet this condition.

Figs. 1, 2, 3, and 4 indicate diagrammatically some modern cylinder arrangements. In current practice the cylinder head is almost always detachable. In Fig. 1 the cylinder block is secured to a crankcase of deep section, the latter providing a very rigid mounting for the crankshaft. While designs with the crankcase joint on the level of the crankshaft axis are common, the

depth of the top half of the crankcase shown in Fig. 1 is a commendable feature. The valves are operated by overhead camshaft, the design being such that the head can be lifted without disturbing the timing chains. This feature facilitates decarbonizing, etc., and is being adopted to an increasing extent in commercial vehicle engines.

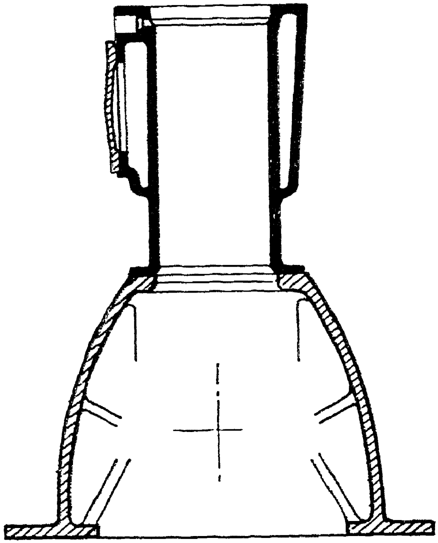


FIG. 1. SEPARATE CYLINDER BLOCK AND CRANKCASE CONSTRUCTION

Fig. 2 shows an arrangement incorporating a separate block and top half of crankcase, and a wet cylinder liner. The rigidity of the assembly is greatly improved by the positioning of the block/crankcase joint some distance above the lower end of the cylinder liner, and the deep section of the crankcase itself.

Since it is not usually possible to pass the big end of

the connecting rod through the bore, this design suffers from the assembly point of view. This is on account of the difficulty encountered in entering the pistons into

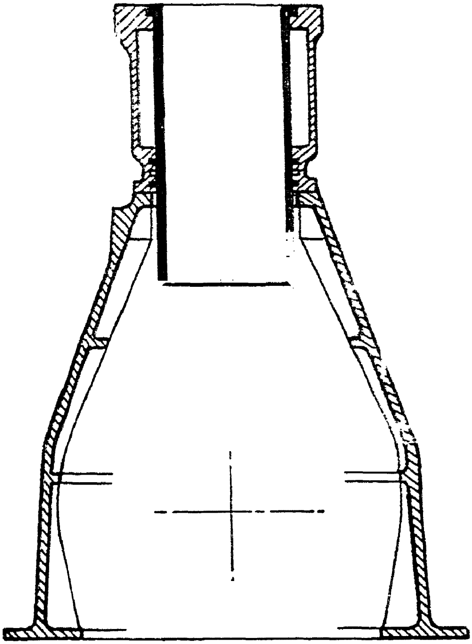


FIG. 2. SEPARATE BLOCK AND CRANKCASE CONSTRUCTION WITH WET LINER

their respective bores as the block is lowered into position.

A modern private car engine design is shown in Fig. 3, the cylinder block and crankcase being combined in a single unit. This arrangement provides a very rigid foundation for the engine, while the one piece casting reduces manufacturing cost. In this design provision

has to be made for removing the pistons and connecting rods downwards from the crankcase, without the necessity of disturbing the crankshaft. The assembly

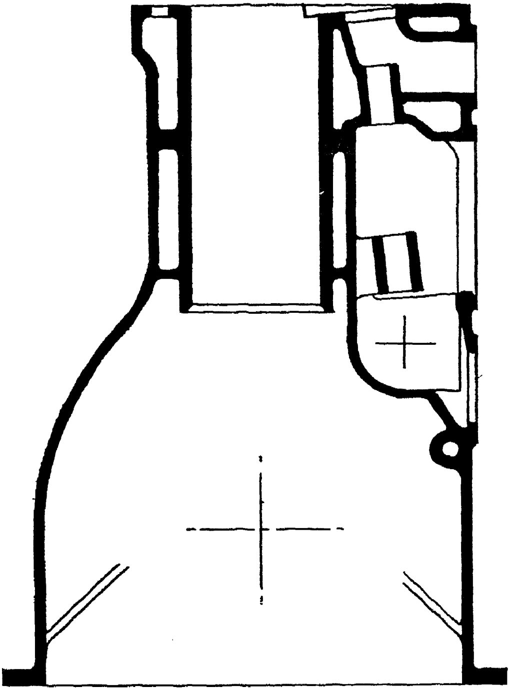


Fig. 3. Combined Cylinder Block and Crankcase Construction

is simpler than that in the arrangements shown in Figs. 1 and 2.

When the crankcase joint is positioned well below the crankshaft axis, the bottom half becomes merely a pan. This may be a pressing or a casting in aluminium or cast iron. The use of a pressing provides a cheap

method of closing the crankcase, but lacks rigidity. Where the bottom half forms a deep sump, it is more usual to specify a casting.

The arrangement shown in Fig. 4 includes a combined cylinder block and crankcase and also a dry cylinder liner. The inserted valve seat will also be observed.

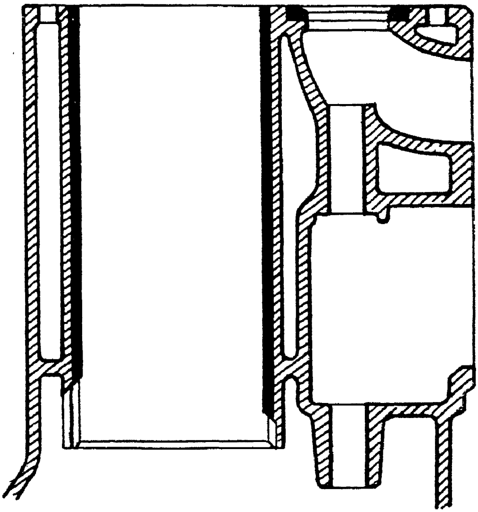


FIG. 4. COMBINED CYLINDER BLOCK AND CRANKCASE WITH DRY LINER AND VALVE SEAT INSERT

Modern operating conditions, particularly with reference to commercial vehicle engines, demand long periods without the necessity for maintenance, and on this account the use of valve seat inserts is increasing. As is the case with renewable cylinder liners, the inserted valve seat renders possible the use of materials, the properties of which ensure longer periods of operation before attention is necessary. The question of a

suitable method of assembly in the cylinder block requires careful attention. Valve seats pressed into recesses in the cylinder block, with an interference fit, tend to work loose under the heating and cooling conditions of operation in service. A better method, which allows more latitude in regard to material specification, is to secure the insert in the cylinder block by means of screw threads or by using some form of spring ring. With renewable liners and valve seats the life of the cylinder block is prolonged almost indefinitely.

Since the crankcase or cylinder block and crankcase combined is virtually the foundation structure of an engine, it is essential that this should be as stiff as possible in the horizontal and vertical planes, consistent with normal weight limitations. It has been known for some time, though probably not always realized, that for smooth operation maximum crankcase rigidity is required. With increased stiffness the natural frequency of vibration of the crankcase is increased, and only the higher harmonics of the periodic forces set up by the reciprocating parts can synchronize with it. As these higher harmonics are of small magnitude their effect is negligible, and vibration or engine roughness is avoided.

This desirable stiffness is usually attained by the provision of adequate ribbing, especially in the neighbourhood of the main bearing supports, by the positioning of the bottom face of the crankcase well below the crankshaft axis, and by the provision of a substantial flange at the bottom face. These features may be observed in the arrangements illustrated in Figs. 1, 2, and 3.

Further methods of increasing crankcase stiffness are depicted in Figs. 5, 6, and 7, these applying particularly to heavy commercial petrol and Diesel engines. Fig. 5 shows an exceptionally heavy bottom

flange of somewhat unusual shape, while Figs. 6 and 7 illustrate the use of box and open girder sections respectively.

Figs. 22 and 23 show further designs incorporating wet and dry liners respectively. The design shown in Fig. 22 embodies a combined block and top half of

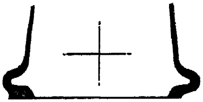


FIG. 5

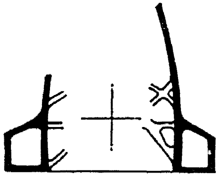


FIG. 6

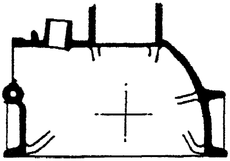


FIG. 7

crankcase, twin camshafts being located high up near the cylinder head joint. Fig. 23 shows a side-valve design, the block being fitted with a dry liner.

The production of V type twelve and sixteen cylinder automobile engines has further complicated cylinder block manufacture. Current practice lies in the direction of having the two banks of cylinders and the crankcase in a single unit, the rocker arm and central camshaft supports holding the two blocks together.

The Cylinder Head. The detachable monobloc cylinder head is almost a universal feature in modern engines. It has, of course, been made possible by the development of the monobloc cylinder casting. The advantages of the detachable cylinder head type of construction include the following—

1. It simplifies the cylinder block casting, providing an open bore. This is of considerable value from a production standpoint.

2. It permits of closer spacing of the valves, thereby reducing cylinder centres.

3. It eliminates the use of valve plugs which are uncooled, and are always a potential source of leakage.

4. It allows of the simple top overhaul, i.e. decarbonization and grinding-in of the valves. This has become an important part of modern engine maintenance.

5. It provides a ready means of adjustment for the compression ratio. If, when tested, the compression pressure is below the limit specified, it can be increased by grinding a small amount off the joint face of the head. The thickness of metal to be removed can be determined by calculation.

The detachable head, however, renders necessary the provision of a gas and water joint over a relatively large area. With good design and careful treatment this disadvantage is not a serious one. The requirements are a suitable copper asbestos joint, an adequate number of holding studs suitably placed, and a sufficient depth of head casting to ensure satisfactory rigidity. Any joint trouble is generally due to lack of rigidity, or insufficient or badly placed studs. The cylinder heads of side valve engines are more likely to suffer from lack of rigidity than those of overhead valve units. In the latter type, the depth of head necessary to accommodate the valves will be sufficient to ensure satisfactory rigidity.

Joint trouble may also be induced by excessive tightening of the holding down nuts. This causes the metal of the block around the studs to be pulled up above the level of the joint face, thereby destroying a great deal of the effectiveness of the joint. Excessive tightening of the holding down nuts may also cause distortion of the cylinder block in the neighbourhood of the valve seats. This results in loss of power, and burning of the valves.

In small engines the cylinder heads are designed in a single casting. In larger engines where the design permits, the heads are sometimes arranged in two castings; this eases the joint problem, and facilitates

the handling of the castings. With the camshaft located in the head, a single casting is, of course, essential. In the case of sleeve valve engines individual cylinder heads are necessary on account of the sealing of the cylinder head extension in the upper end of the sleeve.

Difficulties in the production of cylinder head castings are less acute than in the case of the cylinder block, since the problem of surface hardness does not arise to the same extent. In commercial vehicle engines the practice of fitting renewable valve seat inserts and valve guides is extending; in such designs the life of the head is prolonged indefinitely.

In modern production the combustion chambers are machined all over. This provides better control of the clearance volume and, therefore, the compression pressure.

Cast iron and aluminium are the usual cylinder head materials. As is the case with the specification of aluminium for cylinder block material, the advantages are offset by the increased cost. As the weight is less in the case of the cylinder head, the objection is not so serious. Aluminium is often used in proprietary cylinder head designs for increasing the output of production type automobile engines. The advantages of aluminium in this respect are—

1. It is a more reliable foundry material than cast iron.
2. The cost of machining is less.
3. It provides a certain reduction in weight, although this is not of very great importance in the smaller engines.
4. The higher conductivity of the metal is conducive to better cooling.

A recent development is the use of copper alloys as a cylinder head material. The thermal conductivity

of suitable copper alloys is approximately twice that of aluminium alloys, and six times that of cast iron. In special cases, therefore, where it is required to increase the compression ratio and consequently the power output beyond values normally limited by the tendency to detonation with existing combustion chamber designs, and cast-iron or aluminium cylinder heads, the use of high conductivity copper alloy cylinder heads will provide a solution. In the case of normal engines, however, the higher conductivity material will probably result in over-cooling of the incoming mixture, and increased heat loss generally, thus causing a reduction in thermal efficiency and increased fuel consumption. The application of copper alloy cylinder heads may thus be limited to engines from which the highest possible power output is required, and where fuel consumption is not of paramount importance, and to heavy duty engines normally operating at, or near, full load for long periods, under which conditions the combustion chamber as a whole tends to become overheated.

In normal engines the use of cast-iron cylinder heads with suitably positioned copper alloy inserts may prove advantageous. It will be seen in the discussion which follows on "Combustion Chamber Shape" that the tendency to detonation is affected by the temperature of the last part of the mixture to fire, and that in order to permit the use of the highest possible compression ratio it is necessary that the flame front should travel towards the coolest part of the combustion chamber. The provision of copper inserts in the surface of that part of the cylinder head adjacent to the last portion of the mixture to fire will accelerate heat transfer to the cooling water, and thus provide efficient cooling in such areas.

The iron head is cast round such inserts, which are

finned on the side in contact with the cooling water to facilitate heat transfer.

Cylinder Bore Distortion. One of the many problems which arise in modern automobile engineering is cylinder bore distortion, and its adverse effects on piston and piston ring performance. The modern cylinder block has a complex distribution of metal in its mass, which is variously stressed in the complete assembly, and it has become increasingly apparent that as a result of assembly and operating stresses, the cylinder bores may lose their truly circular shapes both at assembled and at operating temperatures.

Cylinder distortion may be classified as arising from mechanical and thermal causes. Dealing first with thermal distortion: this arises from the effects of heat on the cylinder block structure and the resulting distribution of temperature, and is thus affected by the water-cooling arrangements, such as the length of the water-jackets as compared with the length of the cylinder bores, whether there are water spaces between adjacent bores or not, and the positions of inlet and outlet water pipes.

Mechanical distortion arises out of assembly stresses, and, being most frequently caused by the pulling down of the cylinder head on the studs, is usually present at the upper end of the cylinder bore. It is thus affected by the number and disposition of the studs, and also the thickness and nature of the cylinder head gasket. It may also result in other parts of the bore due to the effects of manifold studs, etc.

Cylinder distortion may take the form of high or low spots, or a general tendency to assume a slightly oval shape with the minor axis along the length of the block. The direct effects of such distortion include interference with piston clearances, resulting either in local seizure or piston slap, "ring flutter" or vibration,

and blow-by. These effects may be minimized by the provision of efficient cooling of the whole periphery, and full length of each bore, and careful disposition of the cylinder head studs.

A recent development has been the use of dummy cylinder heads which are fitted to cylinder blocks while the bores are being machined. The bores are thus finished with the block in the same state of stress as will exist in the assembled engine. Special wrenches, which enable constant tension to be applied to cylinder head studs, are also used.

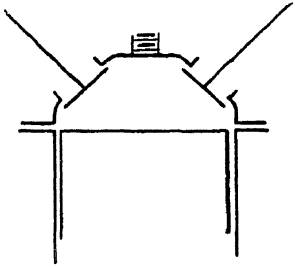


FIG. 8. INCLINED OVERHEAD VALVE HEAD

A further interesting development has been the reverting back to the integral cylinder head and block construction in certain engines of Daimler and Lanchester manufacture. In this way gasket troubles have been eliminated and distortion greatly minimized.

Combustion Chamber Shape. The shape of the combustion chamber is one of the most important features in the design of modern automobile engines. It determines the highest useful compression ratio which may be adopted, and thereby sets a limit to the power and efficiency obtainable. Modern cylinder head design is subject to the following requirements—

1. The shape of the combustion chamber must permit of the employment of a reasonably high compression ratio with freedom from detonation. In present-day practice compression ratios in private car engines range from 5 : 1 to 6 : 1, and round about 5 : 1 in the case of commercial vehicle engines.

2. The shape of the combustion chamber and the nature of the valve location must be conducive to

obtaining a reasonably high volumetric efficiency. Good breathing capacity, especially at the higher speeds, is very desirable in modern engines. The best arrangement for high volumetric efficiency is that shown diagrammatically in Fig. 8, incorporating a hemispherical combustion chamber with inclined overhead valves. The valves open direct into the space above the piston and the entering mixture suffers a minimum change in direction. In modern non-compact combustion chambers, particularly with side valve location, care has to be taken to allow adequate clearance round the valve heads in the avoidance of wire drawing. A thin clearance over a portion of the piston head when the latter is at the top of its stroke is a common feature of such designs, and this results in the formation of a throat between the cylinder bore and the space above the valves. This throat should be designed to create a minimum reduction in the volumetric efficiency.

The so-called "F" head arrangement, shown in Fig. 15, is becoming a popular method of combining a non-compact combustion chamber with high volumetric efficiency. This embodies an overhead inlet valve opening directly into the bore portion of the cylinder head. In this design it is possible to provide a passage connecting all the induction ports.

3. The design, as dictated by the foregoing considerations, must yield a lay-out suitable for modern manufacturing methods, according to the purpose of the design. The valve and sparking plug locations are the factors involved in this requirement. Practically all modern combustion chambers are of the non-compact type, the types of valve location including overhead valves, side valves, and the overhead inlet and side exhaust valve arrangement already referred to.

4. Combustion must be smooth and unobtrusive.

This is a relatively new requirement which has considerably modified previous existing theories, particularly in respect of engines for private cars and passenger vehicles.

Turbulence. The credit for emphasizing the importance of combustion chamber shape must be given to H. R. Ricardo, who initially commenced working on the problem in 1919. It was about that time that the importance of turbulence or mechanical disturbance of the mixture began to be realized. If the mixture is wholly stagnant combustion would not be complete by the time the exhaust valve opened. Turbulence is relied on to speed up the process of combustion. It becomes increasingly important as the density of the mixture is reduced by throttling. Under these conditions the proportion of exhaust gases in the mixture is increased, and since these act as diluents, the flame temperature is lowered, and the process of inflammation retarded. Turbulence also reduces the effective thickness of the cold, relatively stagnant layer of mixture adjacent to the combustion chamber boundaries. As the shape of the head departs from the compact hemispherical form, this inert layer becomes larger and also thicker, and the effect of turbulence is less. In the non-compact combustion chamber with a thin clearance space over a portion of the piston head, the mixture is violently ejected from this space during the final stages of the movement of the piston towards top centre, thereby contributing to the turbulent conditions obtaining during combustion. It will be seen later that the degree of turbulence must be controlled in the interests of smooth running.

Detonation. Ricardo defines detonation as the setting up of an explosive wave or almost instantaneous pressure rise, caused by the further compression of the unburnt portion of the mixture by the very rapid

pressure rise of the burning portion, to a point when the rise in temperature of the unburnt portion is faster than the heat can be transferred. At this point the unburnt portion ignites at one time, causing the familiar high-pitched noise as the explosive wave strikes the cylinder walls.

Apart from compression ratio and the fuel employed, the tendency to detonate depends on—

(*a*) The size and shape of the combustion chamber;

(*b*) The disposition of the valves;

(*c*) The position of the sparking plug.

The location of the plug determines the maximum travel of the flame front to the combustion chamber boundaries, and this requires careful consideration in relation to the temperature and volume conditions of the last portion of the mixture to burn. If the flame travel is long, a large range of ignition will be necessary to ensure combustion being completed by the time the piston reaches a suitable position in the stroke (that corresponding to about 10° after t.d.c.). Under these conditions the head will be very sensitive to ignition advance. The position of the plug should be biased towards the exhaust valve in order to avoid any tendency for the last portion of the mixture to be compressed against the hot exhaust valve head. This factor in minimizing the tendency to detonate was, of course, appreciated before combustion chamber research had progressed very far. Satisfactory ignition at high speeds and when idling are also factors which tend to influence the plug location.

In the early stages of this work it was considered that heat loss was a major factor, and that compact disposition of the combustion chamber volume was contributory to successful design. Such design was thought to minimize heat loss by reducing the surface/volume ratio to a minimum, and by the avoidance of dead

pockets. On the basis of this theory the spherical combustion chamber with centrally positioned plug represented an impossible ideal. The nearest approach to this is the arrangement shown in Fig. 8, which incorporates inclined overhead valves, with the plug positioned on the cylinder axis. The arrangement is practically standard practice in aircraft engines, but the valve gear is somewhat complicated and expensive to general automobile standards. Figs. 9, 10, and 11 show diagrammatically the more familiar forms of combustion chamber in order of their freedom from tendency to detonate. Fig. 9 incorporates the flat cylindrical combustion chamber with vertical overhead valves and horizontally positioned plug. This arrangement is very popular in smaller automobile engines. Fig. 10 shows the orthodox side valve head which is the basis of the modern non-compact combustion chamber. Fig. 11 shows an obsolete arrangement which is very prone to detonation with any plug position.

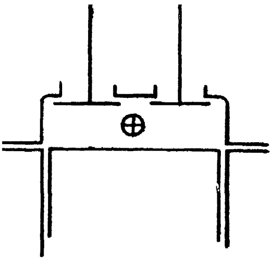


Fig. 9.
Overhead Valve Head

Combustion Chamber Research. In recent years a considerable amount of work on combustion chamber shape has been carried out, particularly in connection with the smooth running factor. This has resulted in the relegation of the heat loss factor to a relatively unimportant position, cylinder head design now being governed by considerations of detonation and turbulence. Outstanding in this connection are the researches of H. R. Ricardo and W. A. Whatmough in this country, and R. N. Janeway in the United States. Although each of these workers has arrived at the

common objective of a smooth and efficient, non-detonating cylinder head, contrary theories have been developed in their published papers in explanation of the results obtained. Since the final result is the same, this disagreement must be apparent rather than real,

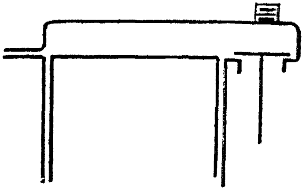


FIG. 10. SIDE VALVE HEAD

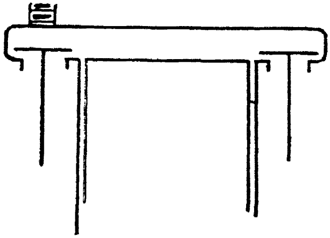


FIG. 11. "T" TYPE HEAD

and examination of their designs has shown that common characteristics exist. According to A. Taub[1] these are as follows.

1. The cooling of the final portion of the mixture to burn by means of a shallow clearance space.
2. The location of the sparking plug at or near the exhaust valve, to ensure that the mixture heated by the exhaust valve will not be the last to burn.
3. The adoption of non-compact combustion chamber shapes.
4. The application of volume control for smooth running by—
   (a) Restricting the initial volume burned.
   (b) Providing maximum flame spread at intermediate inflammation.
   (c) Restricting the flame spread at the end of burning in the main chamber.
5. Acceleration of the rate of pressure rise is the true measure of roughness.

It is interesting to examine the designs and theories of these three experimenters in order to ascertain the degree with which these characteristics apply to each.

Ricardo's Work. The original turbulent head developed by Ricardo for side valve engines in 1919 is

[1] See *S.A.E. Journal*, October, 1930, p. 437.

shown in Fig. 12. Its main features were the provision of a thin clearance above a portion of the piston head, which created additional turbulence during the latter stages of compression, and the reduction to a minimum of the maximum flame travel from the plug to the combustion chamber boundaries. Summarizing his own work in 1929, Ricardo[1] tabulates the objectives of this cylinder head as follows—

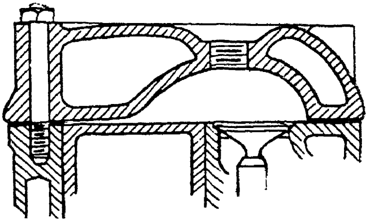


FIG. 12. RICARDO TURBULENT HEAD

(*From "The Automobile Engineer"*)

1. To create additional turbulence during the compression stroke in order to—

(*a*) Increase the rapidity of burning and so obtain both a greater effective expansion ratio and, at the same time, render the engine much less susceptible to ignition timing;

(*b*) Scour away as far as possible the layer of gas which normally clings to the cool faces of the combustion chamber, and is therefore chilled to such an extent as to escape complete combustion, either entirely or until so late in the expansion stroke as to be of little value;

(*c*) Reduce the tendency to detonate by keeping the unburnt gas in rapid motion, thus enabling it the more readily to get rid of the heat of compression by the oncoming flame front and at the same time to break up that flame front.

2. To reduce the length of flame travel from the sparking plug to the farthest point in the combustion chamber by rendering inoperative, so far as detonation is concerned, the part of the combustion chamber over the farthest side of the piston. This was done by the piston coming into such close contact with the head that the gas between these two relatively cool surfaces was so chilled as to avoid detonation.

3. To keep the flame travel as short as possible by placing the sparking plug in a central position.

4. To reduce to a minimum the surface/volume ratio, and

[1] See *Automobile Engineer*, July, 1929, p. 257.

therefore the heat loss during combustion, though this latter is relatively small.

It will be seen from the foregoing that the importance of the temperature conditions of the last portion of the mixture to burn was appreciated. Experiments carried out with a special cylinder head having several quartz windows provided data confirming this. In these experiments the progress of the flame front during combustion was observed by means of a stroboscope over the quartz windows. When detonation occurred a supplementary bright flame appeared at all the windows, but this never occurred until all the windows but the last indicated combustion. Thus it was definitely established that detonation occurs in the last portion of the mixture to burn.

The turbulent cylinder head proved to be too rough in practice. This Ricardo later ascribed to the rate of rise of pressure during combustion. Roughness is defined as a harsh feeling and tendency to drumming, dependent on the beam stiffness of the engine as a whole, and the crankshaft in particular, and on the rate of pressure rise. The necessity for engine rigidity in the interests of satisfactory combustion conditions is a new consideration arising out of combustion chamber research. In the later Ricardo design, known as the "shock-absorber," head smoothness is sought through control of turbulence. In this connection Ricardo[1] points out that—

1. A high degree of turbulence does not increase the maximum pressure, but does influence the rate at which this maximum pressure is attained.
2. Rapid rate of pressure rise, with its subsequent sudden application of pressure, causes springing of the engine mechanism, resulting in roughness. This roughness determines the limit of the degree of turbulence which may be employed.

[1] See *Automobile Engineer*, August, 1929, p. 284.

3. The degree of turbulence which an engine can sustain without roughness is a measure of its rigidity.
4. A well-designed engine will operate smoothly with a rate of pressure rise of 30 lb. per square inch per degree of crank-angle, and only an exceedingly stiff engine will be smooth at 35 lb. per square inch per degree.

From the foregoing it is seen that Ricardo considers maximum pressure to be unimportant as far as roughness is concerned, and that maximum rate of inflammation is the important factor. Turbulence affects only the rate of burning, and it is therefore through controlled turbulence that smooth running must be sought. It is argued that if the initial pressure rise is gradual, a very rapid rate in the later stages of combustion will not induce roughness. To illustrate this the analogy of the silent cam form is used. This embodies a gradual take-up of the backlash, followed by a very high acceleration. The application of this analogy is criticized on the score that as such a cam form may cause surging, so in the cylinder head, where such conditions obtain, the gradual inflammation with accelerated intermediate and final stages will be rough because such latter stages will deflect the mechanism. In other circles constant acceleration of rate of pressure rise is thought to be ideal.

The so-called shock absorber head is shown in Fig. 13. In addition to the thin clearance, this design incorporates a shallow pocket over the exhaust valve containing about 15 per cent of the mixture. The sparking plug is positioned over this pocket. Consideration of this design will show how the principle of volume control for smooth combustion, as further referred to in discussing Janeway's contribution to combustion chamber progress, is applicable to combustion conditions in the initial, intermediate, and final stages. Stagnant conditions are considered to obtain in the

shallow pocket which, by its nature and location relative to the plug, tends to localize and slow down the

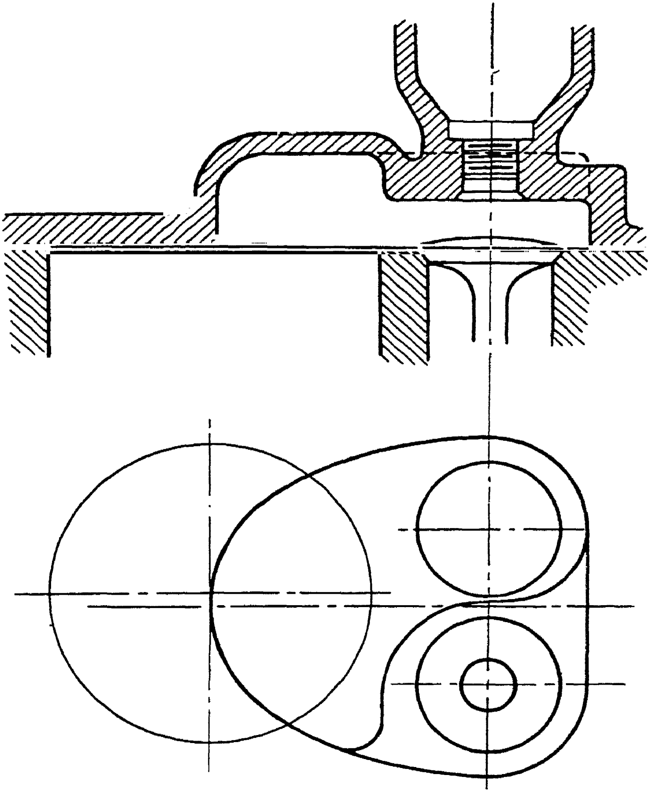


FIG. 13. RICARDO "SHOCK ABSORBER" HEAD
(From "The Automobile Engineer")

commencement of combustion. It will be seen that in the intermediate stages the area of flame front increases rapidly, and finally decreases towards the thin clearance. According to Ricardo the rate of pressure rise

may be as much as 50 lb. per degree with relative smoothness during the later stages of combustion.

Whatmough's Principles and Designs. The principal factors which emerge from an examination of the writings of W. A. Whatmough on combustion chamber design are as follows—

1. Control of heat exchange throughout the working cycle. This includes differential water cooling.
2. Directional firing.
3. Streamlining of the combustion chamber.

In describing his principles of combustion control Whatmough[1] states that—

> Cooling and/or warming of the charge is the one sure guide to that balancing of the many otherwise mysterious flow-factors entering into combustion chamber design. Conclusions deduced from combustion chamber shape will mislead whenever they do not conform with the heat control, which is the one decisive factor in that regulated burning which smooths power output and increases engine efficiency.

This statement is considered to refer to the heating of the initial portion of the mixture to burn, and the cooling of that portion which is being heated by the burning portion. According to Whatmough, considerable quenching of the flame occurs in the thin clearance space, although this is disputed by other authorities.

A typical Whatmough cylinder head is shown in Fig. 14. This is described as anti-turbulent, thereby indicating, in contrast with Ricardo's theories, that low turbulence is necessary for satisfactory combustion. With regard to turbulence Whatmough[2] states—

> Turbulence is the theoretical explanation for increase in flame speed by more or less violent agitation of the combustible during the compression stroke. In practice the "turbulent" L-head permits an increase in compression ratio whereby it develops more power. However, its vagaries in regard to

[1] See *S.A.E. Journal*, September, 1929, p. 250.
[2] See *S.A.E. Journal*, September, 1929, p. 252.

roughness of running discount its supposed anti-knock properties. Undoubtedly the speeding up of inflammation attributed to turbulence is due to turbulent heating or eddying contact of the unburnt charge with heating surfaces.

It appears, therefore, that Whatmough believes that turbulence causes the mixture to pick up heat from the walls, whereas Ricardo relies on turbulence to dissipate heat to the walls. Which of these conditions obtains depends, of course, on the relative temperature of the

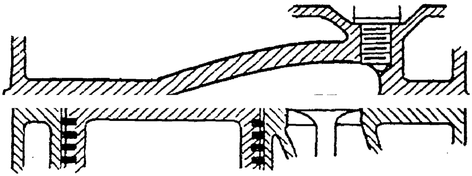


FIG. 14. WHATMOUGH "ANTI-TURBULENT" HEAD
(*From "The Automobile Engineer"*)

mixture and the walls with which it is in contact respectively.

With reference to the general principle of heat control Whatmough aims to consider the heat exchanges operative throughout the whole cycle rather than to those occupying the fraction of it representing the combustion period. It is pointed out that although detonation occupies only $\frac{1}{200}$th part of a cycle, the events leading up to it occur during the whole cycle. Roughness control is aimed at through regulation of the heating and cooling effects in the engine as a whole. Whatmough also stresses efficient distribution and accurate carburation as an important preliminary to good combustion chamber design.

Differential water cooling is part of Whatmough's control mechanism, being employed to regulate the heat exchanges between the mixture and the boundaries of the combustion chamber. The need for water flow

control and forced circulation by deflection round the plug boss and the exhaust valve pocket is recognized.

Directional firing is ignition from "hot" to "cold," and is obtained by the combination of exhaust valve location for the plug and a cooling clearance space. Whatmough's claims[1] for directional firing are as follows—

1. Reduction of initial lag in pressure rise owing to an increase in flame speed due to charge being heated by the hot exhaust valve head.
2. Lowering of peak pressure because the flame is continuously progressing toward the cooling zones.
3. Spreading of higher pressure over a wider working range consequent on delayed burning.

Examination of the design shown in Fig. 14 will reveal how directional firing is incorporated. The plug location past the exhaust valve provides warm initial burning with the flame front travelling towards the cooling clearance space. It will be observed that while Ricardo controls the initial stages of combustion by modifying the height of the chamber in the vicinity of the plug, Whatmough positions the plug at the intersection of two contours. Although Whatmough states that volume control plays no part in Whatmough-Hewitt combustion chambers, the similarity between these designs and those of Ricardo and Janeway is striking in the initial, intermediate, and final stages.

Whatmough head designs indicate a close attention to the provision of adequate volumetric efficiency in the streamlining of the throat, and also round the valves. In the interests of breathing capacity and heating of the charge during induction and compression, turbulent heating of the exhaust valve is avoided. This is particularly the case in the F-head design shown in Fig. 15. The favourable points in this design in relation

[1] See *S.A.E. Journal*, September, 1929, p. 251.

to valve location and high volumetric efficiency have already been referred to.

Janeway's Work. In 1924, as a result of investigation of the Ricardo turbulent cylinder head design, R. N. Janeway[1] arrived at the following conclusions—

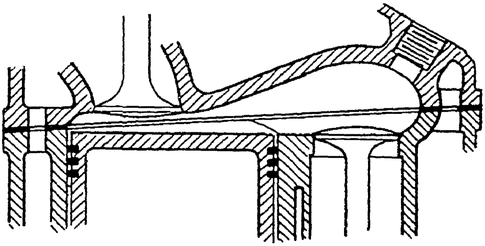


FIG. 15. WHATMOUGH HEAD WITH SIDE EXHAUST AND OVERHEAD INLET VALVES
(From "The Automobile Engineer")

1. Regardless of form, the best firing position for power and detonation is the centre of mass.
2. Combustion chamber form has very little influence on thermal efficiency for any given compression ratio.
3. The offset type of head for L-head engines offers the best possibilities for increasing efficiency through higher allowable compression ratio for no detonation. However, the compactness inherent in the extreme offset chamber tends to produce a harshness.

Regarding the question of plug location Janeway had later to modify this conclusion in the interests of smoothness, the plug position being closer to the exhaust valve, and on the valve centre line. In item (2) Janeway is arguing against Ricardo's claim for higher efficiency due to turbulence. This is confirmed by his experiments with conventional and offset heads, from which he concludes that at the same compression ratio there is no difference in thermal efficiency. Janeway is of the opinion that the Ricardo head is no more

[1] See *S.A.E. Journal,* October, 1930, p. 428.

turbulent than the other type, very little additional turbulence being caused by the throat, if the volumetric efficiency is adequate.

According to Janeway the reason for item (3) is the effect of the clearance space. This effect he has investigated, both in regard to thickness of clearance space and extent of cover of the piston crown. As a result of variation in the latter, detonation is reduced up to 20 per cent cover, and thereafter increases slightly. The slight increase may be due to the inclusion of the hotter portion of the piston crown in the clearance space.

In 1928 Janeway[1] had arrived at the following conclusions—

1. That the effect of the combustion chamber on detonation is almost entirely due to heat transfer affecting the temperature of the part of the charge which burns last and is determined by the following factors: (*a*) location of that portion of the charge which burns last, (*b*) shape of chamber, and (*c*) location of spark plug.
2. That while turbulence is an important factor in engine operation, according to experiments, the combustion chamber form does not appreciably influence the effective turbulence, within limits.

Thus all three investigators are agreed that the last portion of the mixture to burn must be in the coolest place. Since heat transfer is recognized as the important factor, a large variation in chamber shape is permissible, with the same result. Janeway illustrates this principle in a series of diagrams shown in Fig. 16. This is said to have been inspired by Ricardo's emphasis on the heating of the unburnt portion by compression due to the rapidity of inflammation of the burning portion in his definition of detonation. In the turbulent head design Ricardo sought to control this effect by shortening the maximum flame travel, and ascribed the freedom

[1] See *S.A.E. Journal*, October, 1930, p. 431.

from detonation to this. Diagrams *A* to *E* (Fig. 16) represent a combustion chamber divided into ten volumes. The condition at *A* might be the instant of ignition, the volumes being equal. As each volume burns it compresses both the burning and unburnt portions, until at *E* the last portion is very highly

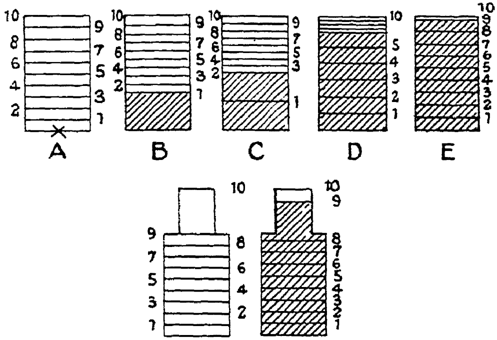


FIG. 16. DIAGRAM SHOWING IMPORTANCE OF TEMPERATURE AND VOLUME CONDITIONS OF LAST PART OF CHARGE TO BURN
(From "The Automobile Engineer")

compressed. In *F* and *G* the clearance space is represented by a section with increased surface/volume ratio, and the improvement in cooling at *G* can be appreciated.

According to Janeway an engine will be rough when sudden changes take place in the rate of pressure rise. This is contrasted with Ricardo's principle of the maximum rate of pressure rise expressed per degree of crank angle, as the important factor. Although the smooth acceleration principle is not acknowledged as a factor in their designs, those of Ricardo and Whatmough have been shown to incorporate low acceleration values. Janeway's method of obtaining smoothness is

by volume control. Taub explains this principle by considering a conical combustion chamber. With the plug

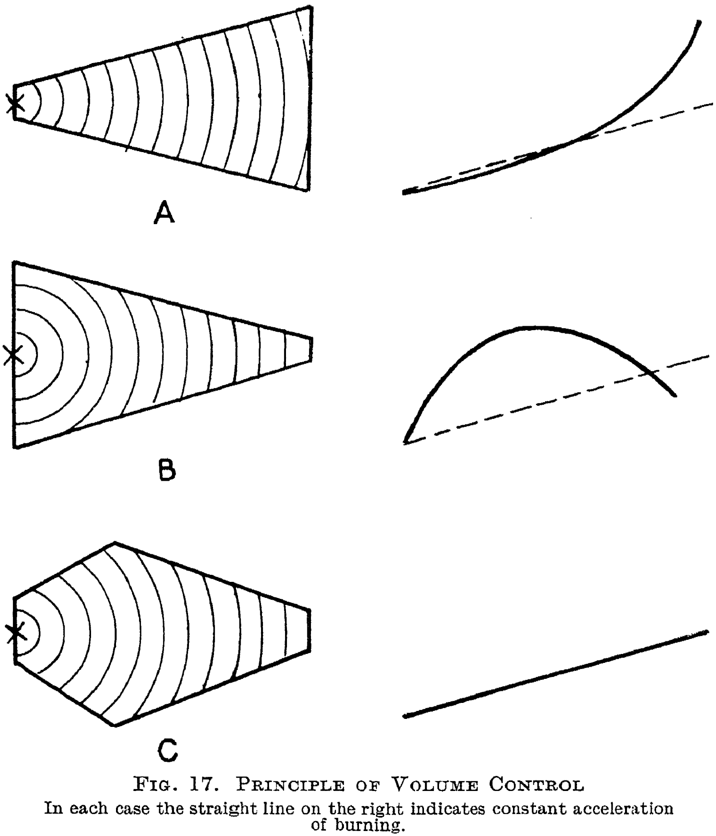


FIG. 17. PRINCIPLE OF VOLUME CONTROL

In each case the straight line on the right indicates constant acceleration of burning.

(*A*) = Increasing acceleration.
(*B*) = Decreasing acceleration.
(*C*) = Constant acceleration.

(*From "The S.A.E. Journal"*)

at the apex the area of flame front progressively increases, while with the plug at the base the flame

front is progressively reduced as the end of burning is reached. In the latter case the final stages would be favourable, but the initial acceleration would be too rapid for smooth results. These two cases are shown in *A* and *B* (Fig. 17). In *C* is shown how the conical

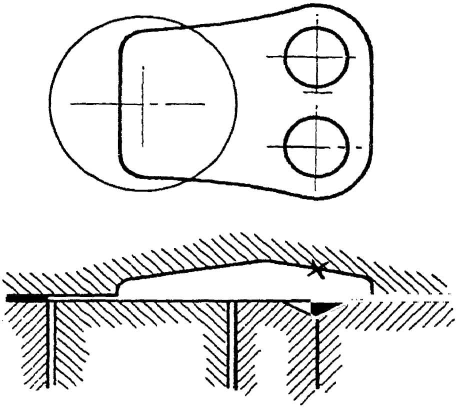


FIG. 18. JANEWAY "ANTI-SHOCK" HEAD
(From "The Automobile Engineer")

shape can be modified to give uniform acceleration throughout inflammation.

Janeway's principles are illustrated in the "anti-shock" head shown in Fig. 18. Both Ricardo and Whatmough designs show controlled initial and intermediate stages. It will be seen that volume control is the common factor in the designs of these investigators. It follows that this principle, which is definite though flexible, makes it possible to arrange the chamber volume around the ignition point to give almost any desired result. Janeway has applied the principle to a mathematical investigation, to permit of the

pre-determination of combustion chamber shape. The relation between ratio of pressure and percentage of total charge burned is worked out for average conditions. The volume of mixture burned at any position of the flame front is calculated assuming uniform progression. The combustion chamber is then divided into flame front volumes and the pressure at each volume burned determined by means of the relative pressure curve. The analysis assumes a constant piston position, but this can be allowed for when the chamber is designed. Velocity is obtained as a function of the pressure and the time element is thus introduced. Acceleration of rate of pressure rise can be analysed, and any condition of excess acceleration due to excess volume, or abrupt change of section at any point, can be corrected. The importance of Janeway's contribution to combustion chamber progress would appear to lie in the practical medium for the determination of chamber shape offered in the principle of volume control.

It will be gathered, however, that the pure mathematical approach to the determination of satisfactory combustion chamber shapes is involved and laborious. An alternative method which is probably more usually adopted by designers and technicians is the checking of proposed designs based on careful consideration and past experience. This type of analysis is described by Mr. G. F. Gibson in a paper entitled "The Four-cylinder Engine,"[1] and reference is also made to a practical fixture which enables corresponding values of flame travel and spherical increment of volume to be easily obtained. A plaster cast of the combustion chamber shape to be analysed is mounted in the fixture, a cutter on which is operated to remove spherical increments of volume, working from the position of the

[1] See *Proc. I.A.E.*, 1936–7, p. 323.

plug electrodes as a centre. Mr. Gibson describes this method as follows—[1]

Experience indicates that the desired essentials to be known are the spherical increment volume expressed as a percentage of total volume, and the flame travel as a percentage of total travel. Having prepared the plaster cast of the combustion chamber, and with the cutting of spherical increments starting from the position of the plug electrodes, the two determined points, i.e. the percentage that each spherical increment is of the total volume, and the percentage that each increment of radius is of the total flame travel are determined; these results are then plotted as shown in the lower curve of Fig. 19, which represents change of volume against flame travel.

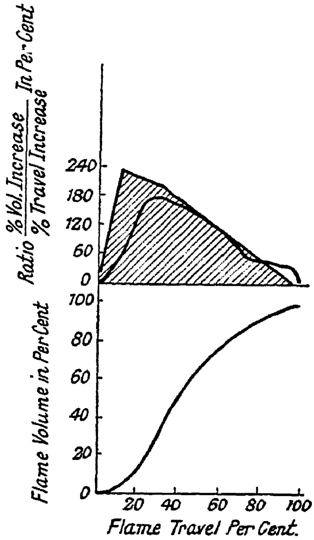


FIG. 19

The upper curve is a derivative of the lower, representing the rate of change of slope of the lower curve, or, in other words, each ordinate of the upper curve represents the slope of the tangent of the lower curve at a corresponding point.

The shaded area on the upper curve represents a limiting area for smooth running, the envelope of which is dictated by experience, since it has been consistently proved that all derived curves lying within this area represent combustion chambers giving satisfactory results; it has also been found that in the interests of smooth running it is advisable for the peak of the upper curves to be within one-third of the total flame distance.

The curve shown in Fig. 19 represents a satisfactory combustion chamber, whilst that of the combustion chamber indicated in Fig. 20 would be judged as unsatisfactory, and this proved to be the case when tested on the road.

[1] See *Proc. I.A.E.*, 1936–7, p. 331.

It will be appreciated that the foregoing mainly concerns the application of combustion chamber research to chambers of the types shown in Figs. 10 and 15, i.e. the side valve and overhead inlet—side exhaust valve arrangements. In the case of overhead valve engines the shape of the combustion chamber is largely dictated by the necessity for inserting the valves.

Application of volume control to an overhead valve engine combustion chamber is to be seen in the design of a recent Chevrolet engine, shown diagrammatically in Fig. 21. In this case it will be seen that the piston crown is of special domed shape, thus attenuating the combustion chamber on the side opposite the sparking plug.

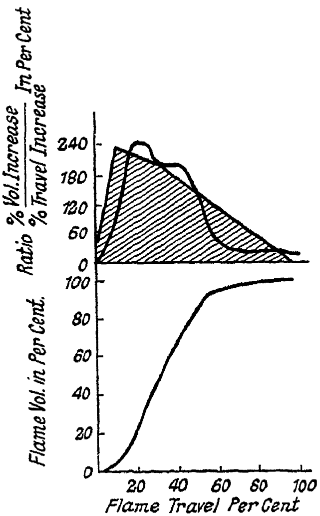


Fig. 20.

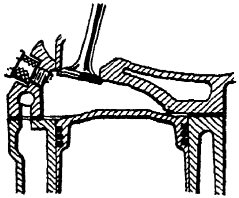


Fig. 21.

Cylinder Liners. Cylinder wear in modern automobile engines, especially those used in commercial vehicles, is a serious problem. In general the importance of renewability of wearing parts is fully recognized by designers. In the case of the cylinder bore, the use of renewable cylinder liners to restore the bore to its original size is now regarded as the only logical solution.

The practice of incorporating cylinder liners in commercial vehicle engines is extending. In many cases the original design embodies this feature, while in others vehicle operators can modify the arrangement and fit liners when the useful life of the original bore has been obtained. In isolated cases the practice has been adopted in engines for private cars. This development has received considerable impetus of recent years by constructional developments which have rendered the foundry problem more acute, and by the increased severity of conditions of operation.

The chief advantage arising out of the adoption of renewable cylinder liners is that their incorporation in the design permits of the separation of the two functions otherwise simultaneously performed by the cylinder block. The choice of the material is therefore simplified. The composition of the material for the cylinder block can be dictated solely by foundry considerations, while in the case of the liner, the specification and method of production can be so arranged as to be conducive to obtaining the most satisfactory bearing surface for the pistons. As a result of this development the manufacture of cylinder liners for supply to engine manufacturers and vehicle operators, as unmachined castings, or as finish-machined liners ready for fitting to the cylinder block, has become a specialized branch of the industry.

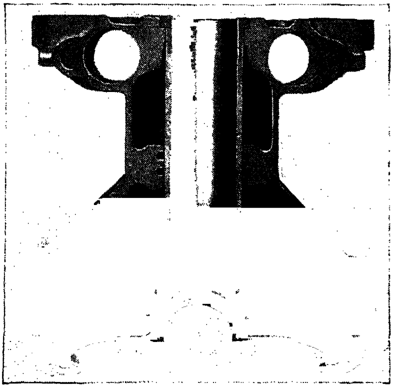
There are two types of cylinder liner. These are designated the "wet" and "dry" types. The dry cylinder liner is considered to be the more important of the two, this being on account of the fact that it can be incorporated either in the original design, or at a later stage when the life of the original cylinder bore has been obtained. It consists of a simple cylindrical sleeve, the inside diameter of which is the specified diameter of the bore, while the wall thickness is as small as will permit

of satisfactory insertion in the cylinder block. This sleeve is ground on the outside, the diameter being dimensioned to provide an interference fit in the portion of the cylinder block bored for its entry. This type of fit must be provided on account of the necessity for good thermal contact between the liner and the supporting wall in the cylinder block, in order to ensure satisfactory cooling of the barrel. The inside diameter of the liner is left slightly smaller than the finished bore diameter, for boring, honing, or grinding after the liner has been pressed home. To facilitate fitting, a "lead" is usually provided at the lower end of the liner. A side valve engine cylinder block with a dry liner fitted is shown in Fig. 23. In the illustration, one of the liners is shown at an intermediate position in the pressing operation. In some designs the liner has a flange which locates in a recess at the cylinder head end of the block. When cylinder liners are fitted by the vehicle operator, the worn bore requires to be machined out to the diameter necessary to accommodate the liner.

The fitting of dry liners may also be facilitated by cooling same in a refrigerator chamber, using liquid air, solid carbon dioxide, or some other refrigerant. In this way the interference fit is absorbed by the resulting contraction in the liner diameter, and liners so handled can be fitted without pressure.

The wet type of cylinder liner is shown in the arrangements shown in Figs. 2 and 22. It will be appreciated that the wet liner must be arranged for in the original design. In this type of design there are no barrels cast integral in the cylinder block, the latter consisting essentially of a water jacket casting designed to carry the cylinder head, and to bolt on to the crankcase. The liner in this case forms a complete cylinder bore, the outside of which is in direct contact with the cooling

water. The liner, which is flanged at the upper end, is supported in the block adjacent to the cylinder head face, and at a point near the lower end of the liner. Since the liner is in direct contact with the cooling water, special jointing arrangements must be employed to ensure water-tight joints at the top and bottom of



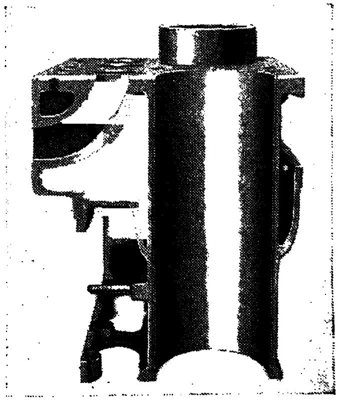


FIG. 22. WET LINER ASSEMBLY FIG. 23. DRY LINER ASSEMBLY
(British Piston Ring Co., Ltd.)

the liner, in order to prevent any leakage of water into the cylinder head or the crankcase. At the top end the joint is formed by the flange seated in its recess. At the lower end it is usual to incorporate water seal grooves. In the design shown in Fig. 2 three grooves are provided, the top and bottom grooves being fitted with joint rings of special composition. The centre groove is left empty, drain holes being provided to lead away any water leaking past the top joint. An alternative method is to compress a packing ring between shoulders on the block casting and the liner. Since it is not possible to provide a suitable shoulder

on the liner at this point, the same effect is usually obtained by reducing the outside diameter of the liner from this point down to the lower end. Against the small shoulder thus formed, a suitable washer is located.

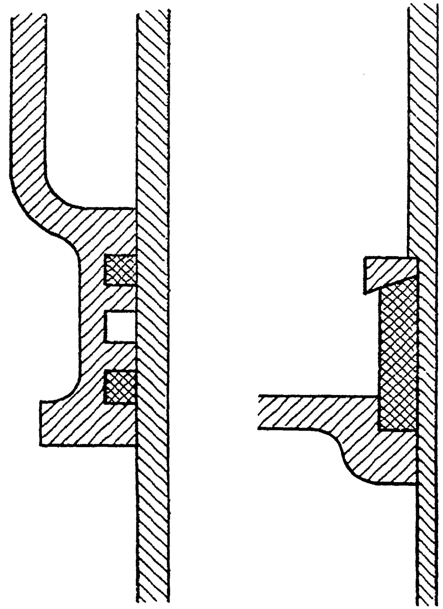


Fig. 24. Methods of Sealing Water Jacket from Crankcase in Wet Liner Assembly

These two methods of sealing off the water jacket from the crankcase are illustrated in Fig. 24.

Apart from the advantage possessed by the dry type of cylinder liner in that existing designs can be modified to accommodate this feature while the wet type has to be allowed for in the original design, the wet type

possesses several advantages over the dry type. These include the following—

1. The cylinder block is greatly simplified since it becomes merely a casing to provide walls for the water jackets, and support for the cylinder heads and liners. The absence of cylinder bores requiring machining eliminates the possibility of blocks having to be "scrapped" after machining operations have been performed on them.

2. The wet liner construction ensures better cooling of the cylinder barrel, the wall section being uniform and of the designed thickness.

3. The wet liner can be finish-machined before fitting to the cylinder block. As has already been mentioned, in the interests of good thermal contact between the dry liner and the supporting cylinder barrel, an interference fit has to be provided. On this account the bore of the dry type of liner must be finished *in situ*. This is not the case with the wet liner, which is given at most only a very slight interference fit in the supporting rings.

The increased use of cylinder liners has led to improvements in the centrifugal method of casting. Liners cast in this way have a close grain and high surface hardness. In addition the use of cylinder liners lends itself to the specification of material which can be heat treated, thereby rendering a further increase in surface hardness possible. In the centrifugal casting process the molten metal is poured into a spinning mould. The centrifugal force due to the speed of rotation throws the material to the walls of the mould and the casting cools while rotating. Any impurities are left at the centre of the casting, and are removed during the casting process. The thickness of the casting is regulated by a plate fitting on the end of the mould. This plate has a hole in it equal in diameter to

the bore of the liner casting required. When sufficiently cooled the casting is forced out of the mould by a plunger. The peripheral speed producing the best type close grain casting is in the neighbourhood of 1,800 ft. per minute.

In the early stages of development, certain difficulties were experienced in producing satisfactory castings by the centrifugal process. A prominent defect was the presence of pinholes caused by gases trapped in the molten metal. This and other difficulties have been overcome by careful control of the casting conditions, the moulds being worked under close pyrometric control. Castings have to be removed from the mould before a chill has time to form, and annealing and tempering are not necessary in the ordinary way.

The usual practice of large operators is to obtain two lives from each liner, the bore being increased by boring and honing to the standard sizes. This simplifies maintenance procedure, and calls for two standards of pistons and piston rings only.

Cylinder Wear. The problem of undue cylinder wear and its causes has exercised the minds of automobile engineers for many years. A point which puzzled investigators for a long time was the inability to reproduce road wear values in engines run on the test bed. In 1932 the Research Department of the Institution of Automobile Engineers initiated an investigation of the problem of cylinder wear, and in an interim report published in 1933 drew attention to an entirely new factor, i.e. corrosion.

In the first place it was found by experiment that abnormal wear could not be induced in the single cylinder test engine under normally varying conditions of load, speed, cylinder bore lubrication, crankcase dilution, and cylinder wall temperature. It was then found that a large increase in wear resulted when the

engine was operated continuously with cold jackets, and a resulting cylinder wall temperature of less than 80° C., and also with a cycle of conditions consisting of stopping, idling, and full load running. Maximum wear was obtained with this cycle of conditions in which delayed warming up was accompanied with delay in the supply of lubricant to the cylinder bore.

The combustion of petrol produces water and acids, and it was held that with the low cylinder wall temperature, condensation of acidic water on the cylinder bore causes corrosion, and that the abnormal wear obtained under low temperature conditions was caused by this corrosion. This conclusion was confirmed by similar low temperature tests with hydrogen, the products from the combustion of which are practically non-acidic, and with this fuel greatly reduced wear values were obtained.

Under conditions of operation at normal engine temperatures, however, cylinder wear still takes place, and the total wear under any conditions can be said to be a combination of corrosive wear and what may be termed "abrasive" or mechanical wear. Corrosive wear takes place during the warming up stages, and its degree is affected by the degree of cylinder bore lubrication, and the delay before which this is fully effective on starting up. Abrasive wear takes place under all conditions of operation, and is dependent on many factors, including the degree of smoothness of the cylinder bores and piston rings, piston and piston ring clearances, cylinder bore and/or piston distortion, ring flutter and blow-by.

The wear in cylinder bores is practically limited to the depth of the piston ring track with the piston at top centre, where it is a maximum, decreasing from there to the lowest point of the piston travel. At a

distance equal to one-half of the piston travel, the wear is generally about one-third of the maximum.

From the foregoing it will be appreciated that cylinder wear may vary over a wide range of values, depending largely on the conditions of operation. Under the same conditions as regards material and lubrication, etc., an engine which operates for most of the time at or near full load temperatures will develop considerably less cylinder wear than an engine which is constantly stopping and starting, and with only intermittent running at part loads, as in the case of local delivery work, etc. For this latter type of service special cylinder liner and piston ring materials with great resistance to corrosion and abrasion are essential if reasonable life is to be obtained. In addition, such engines are kept as warm as possible by the use of radiator shutters, a thermostat in the water system, the elimination of the fan, and the use of cast-iron pistons (on account of low heat conductivity).

In Table I are tabulated Brinell hardness numbers and average values for the mileage per one-thousandth

TABLE I

BRINELL HARDNESS AND WEAR

Part	Material	Brinell No.	Mileage per $\frac{1}{1000}$ in. Wear
Cylinder block and head	Straight-run cast iron	180/220	3,000
	Nickel-chromium alloy iron	220/240	4,000
	Chromium alloy iron	200/220	4,500
Centrifugally-cast liners	Nickel-chromium alloy iron	220/240	7,500
	Special hard nickel alloy iron	400	20,000
	Heat-treatable nickel alloy iron	500	20,000
	Nitrided steel	900/1000	Over 20,000

of an inch wear. The difficulties encountered in obtaining strictly comparable values for wear under similar

conditions of lubrication and service are many, and while the values given in Table I may be taken as average results, they are not comparable in that sense. These values are representative of normal conditions of operation resulting in more or less usual combinations of corrosive and abrasive wear.

In Table II comparative wear figures are given for ordinary cylinder block material and austenitic cylinder liners (referred to under "Materials") for various types of engine and conditions of service.

TABLE II

Type of Engine	Mileage per 1/1,000 in. Wear	
	Ordinary Cylinder Block	Austenitic Liner
Car . . .	3,000	10,000
Passenger Vehicle .	4,000	10,000
Commercial Vehicle .	2,000	8,000
Commercial Vehicle (delivery work) .	300	3,000

Materials. The advancement of automobile engine design which has taken place during recent years, and the more intensive service to which automobile engines are now being put, have given rise to many problems, not the least of which is the necessity for a better cylinder block or liner material. The importance of the material employed for the cylinder bore is increased by the low oil consumption figures demanded from modern engines. Under the resulting conditions of lubrication, experience indicates that the lubrication of the upper parts of the cylinder walls is insufficient to prevent abrasion due to metallic contact. This appears to be particularly the case when scraper rings are fitted

to the piston skirt. The available materials may be classified as follows—

1. Straight run cast irons.
2. Cast irons to which have been added proportions of nickel or chromium or both.
3. Centrifugally cast cylinder liners to either of the above specifications, as cast, or in the hardened state.
4. Austenitic or stainless iron liners.
5. Steel liners, including those which have been subjected to the nitriding process of hardening.

The principal difficulties encountered in the production and use of straight-run iron castings are due to the variations in structure produced by variations in section thickness. Although primarily determined by the composition, the nature of the structure is largely influenced by the rate of cooling, which is directly related to the section thickness. Thus the structure in a single casting may vary from being hard and brittle in thin sections, to an open-grained and weak iron in thick sections. Rapid cooling at corners and edges produces hard spots which reduce machinability and may possibly cause damage to production tools.

In addition, considerable losses due to scrap may be caused by porous patches shown by the water pressure test. Such patches are generally associated with weak and soft metal of poor wearing quality.

In the production of cylinder block and cylinder head castings, cast iron is therefore proving unequal to the demands made of it by modern operating conditions. This applies in particular to its resistance to wear, which, although good, is inferior to that of other materials which have been adopted in consequence. For the purpose under consideration the qualities of cast iron are considerably improved by small additions of chromium or nickel. These improvements consist of the equalization of the structure, thereby reducing

the variations otherwise likely to occur between thin and thick sections, improvement in machinability, and the elimination of porosity. The proportion of nickel added varies between 1 and 2 per cent. Since nickel acts in the same way as silicon in reducing chill, its addition should be accompanied by a reduction in the silicon content of about 0·3 to 0·5 per cent. If this is not possible the same result may be obtained by the addition of a small proportion of chromium. Additions of chromium provide further improvement in the resistance to wear. Compositions of typical cylinder casting materials are given in Table III.

TABLE III

CYLINDER CASTING MATERIALS

"*Chromidium*"—*Chromium Alloy Iron*

(Specification of The Midland Motor Cylinder Co., Ltd., Smethwick)

Total Carbon .	3·10 to 3·30%	Manganese .	0·75 to 1·00%
Combined Carbon	0·60 to 0·80%	Sulphur . .	0·09%
Silicon . .	2·00 to 2·20%	Chromium .	0·25 to 0·45%
Phosphorus .	0·15 to 0·19		

Nickel-chromium Alloy Iron

Total Carbon .	3·00 to 3·50%	Manganese .	0·60 to 1·00%
Combined Carbon	0·50 to 0·80%	Sulphur . .	0·12% (max.)
Silicon . .	1·20 to 1·80%	Nickel . .	1·00 to 1·25%
Phosphorus .	0·04% (max.)	Chromium .	0·30 to 0·50%

Operating conditions have shown that the best results are obtained with centrifugally cast alloy iron liners. This is no doubt due to the close-grained structure resulting from the method of manufacture. The resistance to wear possessed by this type of liner has been shown to particular advantage under the modern conditions of imperfect lubrication already referred to. Strictly comparable tests on cylinder blocks, and centrifugally cast cylinder liners, the material specification of which was the same in both cases, have shown an improvement approaching 50 per cent. Reference to Table I will show that with the

centrifugally cast nickel chromium alloy liner the mileage corresponding to one-thousandth of an inch wear is about 7,500.

Additions of 1 to 2 per cent of nickel give a fine structure, pearlitic in nature. The addition of larger proportions of nickel offers further alternative materials. The harder and denser structure associated with the higher carbon steels, such as tool steel, can be obtained in iron castings by the addition of 5 to 7 per cent of nickel. The structure obtained is that known as martensitic, the resulting castings being tough and hard as cast, with the Brinell hardness about 400. This type of casting is readily machinable with "Widia" type cutting tools, and is finished by grinding.

The necessity for special tools in the machining of the type of casting mentioned above may be looked upon as a certain disadvantage. The addition of a smaller percentage of nickel, usually about 3 per cent, with or without 1 per cent of chromium, provides a casting in which the martensitic structure can be induced by heat treatment. Such castings are relatively soft as cast, thereby simplifying machining, and can be hardened without danger of fracture by quenching in oil or air, giving high hardness and good resistance to wear. The heat treatment of such castings consists of heating to 850° C., and rapidly cooling in air or quenching in oil.

Suggested material specifications for these two types of alloy iron are given in Table IV.

A further type of alloy iron which is of great interest in modern automobile engineering is that known as austenitic or stainless iron. This iron has a composition which includes 14 per cent to 15 per cent nickel, with 7 per cent copper. This material is used with the structure in the austenitic condition, and its special properties include a high coefficient of expansion. This

TABLE IV

CENTRIFUGALLY CAST LINER MATERIALS

"Brico" Nickel-chromium Alloy Iron

(Specifications of The British Piston Ring Co., Ltd., Coventry.)

Total Carbon .	3·5% (max.)	Sulphur . .	0·12% (max.)
Combined Carbon	0·45 to 0·80%	Phosphorus .	0·80% (max.)
Silicon . .	1·80 to 2·50%	Manganese .	0·40 to 1·20%

	Nickel	*Chromium*
Alloy No. 1	0·1 to 0·5%	0·1 to 0·2%
Alloy No. 6[1]	0·4 to 0·8%	0·4 to 0·8%

[1] In Alloy No. 6, Combined Carbon = 0·9% (max.).

Nickel Alloy Iron (Martensitic)

Total Carbon .	3·2 to 3·5%	Phosphorus .	0·5% (max.)
Silicon . .	1·5%	Sulphur . .	0·12% (max.)
Manganese. .	1·0%	Nickel . .	5·0 to 6·0%

Nickel-chromium Alloy Iron (Heat Treatable)

Total Carbon .	3·2 to 3·5%	Sulphur . .	0·12% (max.)
Silicon . .	1·7%	Nickel . .	3·0%
Manganese. .	1·0%	Chromium .	1·0%
Phosphorus .	0·5% (max.)		

is about 50 per cent greater than that possessed by ordinary cast iron, and approaches that of aluminium and magnesium base alloys. This alloy offers a possible solution to the problem of piston slap caused by excessive clearances, and the differential expansion of the piston and cylinder material.

Austenitic cylinder liners also possess the most important characteristic of great resistance to corrosion as well as abrasion. According to tests made with various corrosive media, the resistance to corrosion of this material in dilute acids is some hundred times greater than that of ordinary cast iron. This material may thus be applied with great advantage in engines which operate under conditions resulting in a considerable degree of corrosive wear. Austenitic liners are relatively expensive, but the results obtained with

same under extreme conditions more than balance the initial cost.

For the most satisfactory results it is essential that the piston rings as well as the cylinder liners be made of stainless iron.

The application of carbon steel as a liner material has been the subject of experiment, but has met with no success in the automobile industry. A source of trouble arises from the tendency to "picking-up" with aluminium pistons having close clearances. Where larger clearances are the rule, as in aircraft engines, these liners have been extensively used.

The development of the nitriding process of hardening has provided a further alternative material. Nitrided cylinder liners are now available and are giving exceptional results in service.

The improved results obtained with hardened steel liners must be offset by the increased cost especially in connection with the machining of the bore to obtain a second life. With hardened material a grinding operation is, of course, necessary.

Valve seat wear, particularly in commercial vehicle engines is of almost equal importance as cylinder wear. As already mentioned, the use of valve seat inserts is increasing, and these are now being produced as centrifugal castings. Here again nickel chromium alloy iron has been found to produce very favourable results. Where an interference fit is relied on in assembling the insert in the cylinder block, the special austenitic high expansion alloy should be useful in combating the tendency to working loose.

SECTION V

MECHANICS OF A MOVING VEHICLE

BY

H. KERR THOMAS, M.I.MECH.E., M.I.A.E.

SECTION V

MECHANICS OF A MOVING VEHICLE

Tractive Resistance. The power developed by the engine of a motor vehicle is absorbed in the following way: (1) By friction in the internal mechanical parts of the chassis; (2) by the rolling resistance of the wheels upon the road; (3) by surmounting hills, etc., and (4) by the resistance of the air. The internal friction means that the net power available at the wheels must always be something less than the measured brake horse-power of the engine. The friction in the gear box, for example, varies with the gear engaged, and is due, not only to the sliding friction between the gear teeth and the friction of the various bearings, but frequently in an even greater measure to the churning of the oil in the gear box. Thus, even in an ordinary sliding gear box, when, in top gear, the power of the engine is seemingly passing straight through the shafts to the rear axle, some of it is being intercepted and left behind in the form of heat due to the violent agitation of the oil; and the greater the space round the gears, and the thinner the oil, the less will be the friction. The same argument applies to the rear axle, and in a lesser degree to the universal joints. It is not necessary to go fully into this question, but it is sufficient to say that under the best conditions of top gear performance about 90 per cent of the engine power will reach the back wheels, while on lower gears an additional 7 to 10 per cent will be lost in internal friction, leaving, say, 80 per cent of the total power to reach the driving wheels.

Throughout this section we shall, as far as possible, disregard the term horse-power, and express this in terms of the engine torque in pounds-feet. Thus, Torque $= T = \dfrac{5252 \text{ b.h.p.}}{n} =$ lb.-ft. where $n =$ r.p.m. of the engine; if we put e for transmission efficiency between engine and wheels, and r for the radius of the driving wheels in feet, it is clear that torque at the tyre surface will be $T_t = T \times e$ lb.-ft., and the driving force P in lb. at the tyre is, neglecting gear ratio,

$$P = \frac{T_t}{r} = T \times \frac{e}{r}$$

Suppose we have the following conditions—

$$\begin{aligned} \text{Engine} &= 57 \text{ b.h.p.} \\ \text{r.p.m.} &= 3000 \\ e \text{ on top gear} &= 90 \text{ per cent} = \cdot 9 \\ r &= 1{\cdot}41 \text{ ft.} \end{aligned}$$

then $$T = \frac{5252 \times 57}{3000} = 100 \text{ lb.-ft.}$$

$P = \dfrac{Te}{r} = \dfrac{100 \times \cdot 9}{1{\cdot}41} = 63{\cdot}8$ lb., and on top gear this is the force available for propelling the vehicle.

Tractive Factor. If we divide the torque at the tyres by the total weight of the vehicle, we shall obtain a ratio $\dfrac{T_t}{W}$, which is called the tractive factor of the vehicle, and it is evident that every vehicle has as many tractive factors as it has "speeds" or different gear ratios. It is a characteristic of all internal combustion engines that the maximum torque of the engine is delivered at something less than the speed of its maximum power, and Fig. 3 is a typical horse-power

curve with another curve showing the corresponding torque in pounds-feet at various speeds, as set out in Table I; it is seen that the maximum torque is delivered at about 1,000 r.p.m., and the tractive factor is always

TABLE I

BRAKE HORSE-POWER AND POUNDS-FEET TORQUE

R.P.M.	Torque	B.H.P.
500	203	19·3
1,000	209	39·8
1,500	204	58·2
2,000	186	70·9
2,500	162	77·0

measured at the point of maximum torque for estimation of hill-climbing ability. For a racing car it may, of course, be measured at the point of maximum power,

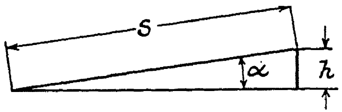


FIG. 1. SHOWING MEASUREMENT OF GRADIENT

or at whatever engine speed the car may be expected to attain.

To propel the vehicle we have to overcome three opposing forces—

1. The rolling friction or road resistance.
2. The force of gravity if ascending a hill.
3. The resistance of the air.

In addition, we must have some reserve of power available for acceleration.

In an ordinary vehicle the most formidable of the opposing forces is that of gravity, and we are thrown

back on the simple problem of the inclined plane, which, in driving, we more usually describe as a gradient, and when we speak of a gradient of 1 in 7 we mean that in passing over a surface distance s (Fig. 1) of 7 ft., we ascend or descend a height of 1 ft. Note that the distance is not measured horizontally.

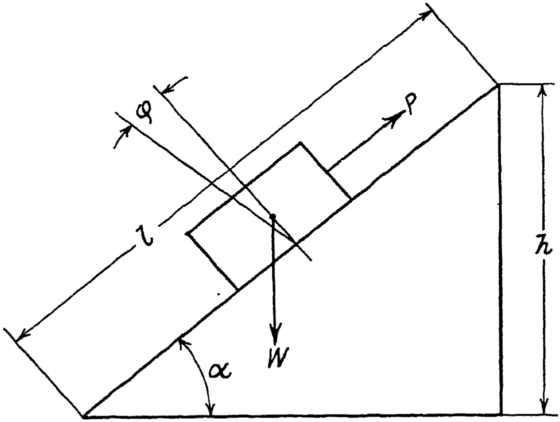


FIG. 2. FORCES ON INCLINED PLANE

Now 1 in 7, or $\frac{h}{s}$ is the sine of the angle of the gradient, so if we have a body W being pulled up the slope, and there is no friction, the force required

$$P = W \sin \alpha \quad . \quad . \quad . \quad (1)$$

or

$$P = W \times \frac{h}{s} \quad (2)$$

and as h in this equation is always equal to 1 we can write $P = \frac{W}{s}$ where in this sense $s =$ the number of times h is contained in s. For greater clearness we will make $G =$ gradient $= \frac{h}{s}$, then if $\frac{h}{s}$ stands for a gradient

of 1 in 4 or 1 in 7 we shall understand that $G = 4$ or 7 respectively, and note that a gradient of 1 in 1 is a vertical ascent, and that then $P = W$, since the sine of $90° = 1$.

The rolling friction and gravity operate in conjunction, and may be dealt with as in Fig. 2, where the

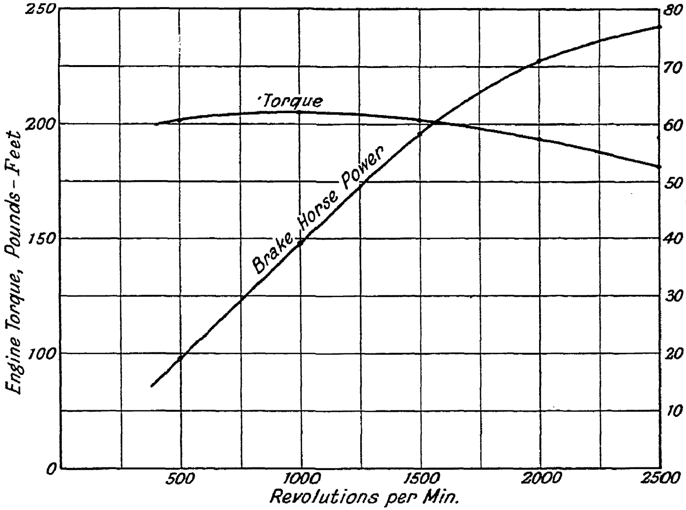


FIG. 3

rolling friction is expressed as $\tan \varphi$, where φ is the angle of repose. Then, taking friction into account,

$$P = W \frac{\sin (\alpha + \varphi)}{\cos \varphi} \quad . \quad . \quad . \quad (3)$$

This equation is the basis of any investigation of the behaviour of a vehicle ascending a hill, but in its trigonometrical form is not capable of immediate application to such data as are commonly available, and equation (2) more generally applies, and we have only to divide the total weight of the vehicle by the

unit of gradient to find the force necessary to propel it up that gradient if the rolling friction is neglected.

For example, if the total vehicle weight $= W =$ 4,000 lb., and it is required to ascend a gradient of 1 in 4, then $P = \frac{W}{G} = \frac{4000}{4} = 1000$ lb., and we must provide this torque *at the wheel* either by the direct power of the engine, or by employing an engine of smaller torque and multiplying it by some ratio of gearing. In the engine chosen (Fig. 3) the maximum torque is 209 lb., and at 80 per cent mechanical efficiency this becomes $209 \times \cdot 8 = 167\cdot 2$, then $\frac{1000}{167\cdot 2} = 5\cdot 98$ and this would be the total gear ratio required for such conditions, *neglecting road resistance.*

In practice, the road resistance is a very considerable

TABLE II

ROAD RESISTANCE IN POUNDS PER TON AND PERCENTAGE OF WEIGHT, FOR PNEUMATIC AND FOR SOLID RUBBER TYRES

Surface	Solid Tyre		Pneumatic Tyre	
	Lb. per Ton	Per Cent	Lb. per Ton	Per Cent
Polished marble	12·125	·541	8·08	·360
Concrete (trowelled)	14·25	·636	9·50	·423
Asphalt	15·40	·687	10·25	·457
Stone sets (good)	16·40	·732	10·92	·487
,, (bad)	35·00	1·562	23·30	1·040
Vitrified brick	19·45	·868	12·95	·578
Good macadam	33·40	1·491	22·20	·995
Fair ,,	50·00	2·232	32·30	1·485
Rough ,,	70·00	3·125	46·60	2·070
Clay	100·00	4·464	66·60	2·972
Sand	300·00	13·400	200·00	8·920

item, as it may be anything from 15 to 500 lb. per ton weight of the vehicle, according to the nature of the road surface, and this resistance has to be added to the resistance offered by gravity. Table II gives some typical road resistances.

These figures must be considered as approximate only, and they show that for normal running a resistance of about 50 lb. per ton may be met with on ordinary roads, equivalent to 2·24 per cent of the weight of the vehicle, and this resistance has to be added to the resistance of the gradient.

The gradient resistance may also be expressed as a percentage of the vehicle weight, since $P = \frac{W}{G}$ as shown in the following table—

TABLE III

SHOWING GRADIENT RESISTANCE AND ROAD RESISTANCE AS PERCENTAGE OF VEHICLE WEIGHT

Gradient G	Gradient Resistance % of W	Road Resistance per Ton	% of W
		Lb.	
1 in 1	100	5	·223
2	50	10	·446
4	25	15	·670
6	16·6	20	·890
8	12·5	25	1·115
10	10·0	30	1·339
12	8·3	40	1·785
14	7·1	50	2·230
16	6·2	60	2·680
18	5·5	70	3·12
20	5·0	80	3·57
22	4·55	90	4·02
24	4·17	100	4·46
26	3·85	125	5·59
28	3·57	150	6·70
30	3·33	175	7·82

The figures above are plotted as a chart in Figs. 4 and 5, from which any intermediate values can be obtained.

Air Resistance. We next come to the consideration of the air resistance: this opposing force is dependent upon the speed of the vehicle, and increases with the square of its velocity. For a flat plane moving in a direction perpendicular to its surface, the air resistance is kV^2A. Where V is the velocity in feet per second and A the area in square feet the constant k may vary through wide limits with the shape of the car, and any figure given can only be an approximation, since a road vehicle may vary in shape from a square-fronted object, like a covered lorry, to a stream-lined affair shaped like a cigar. Only direct experiments in an aerodynamical wind tunnel can determine the precise value of the constant for any particular car, but it will be sufficient for our purpose if we take as an approximate value 0·0017 for an average motor car of modern design and 0·0006 for a fully stream-lined racing car, and 0·0024 for a lorry or omnibus. These figures are probably on the low side. The air resistance is also much dependent on the force of the wind and its direction; thus, if a car is travelling against a head wind, blowing at 20 m.p.h., the speed of the car over the road being 30 m.p.h., the virtual air velocity will be $30 + 20 = 50$ m.p.h. If the same wind were *following*, the virtual air velocity would be $30 - 20 = 10$ m.p.h., while if there is no wind the air pressure will be that due to 30 m.p.h.

The following table shows the wind pressure for the three forms of body at various speeds. It is seen that the square-ended vehicle is not sensibly impeded by air resistance until its virtual velocity approaches 20 m.p.h., and below this speed the effect of air resistance may safely be neglected. In the case of the touring

car the resistance is perceptible at about 25 m.p.h. and with the racing car at about 45 m.p.h.

Unlike the other two forces we have considered, the air resistance has nothing to do with the weight of the car, and must be added to the others separately.

TABLE IV

SHOWING WIND PRESSURE LB. PER SQUARE FOOT OF EXPOSED AREA FOR THREE TYPES OF VEHICLE AT VARIOUS SPEEDS (*Approximate Values*)

M.P.H.	V; ft. per sec.	$\cdot 0024 V2$	$\cdot 0017 V^2$	$\cdot 0006 V^2$
10	14·67	·516	·366	
20	29·35	2·06	1·46	·516
30	44·00	4·65	3·29	1·16
40	58·60	8·24	5·84	2·06
50	73·30	12·90	9·13	3·22
60	88·00	18·60	13·16	4·65
70	102·60		17·90	6·32
80	117·30		23·40	8·26
90	132·00		29·65	10·45
100	146·70			12·92
150	220·00			29·00

Summarizing these results : (1) The resistances to propulsion, or, as they are generally known, the "Tractive Resistances," are : The gradient resistance, the surface or rolling resistance, and the air resistance. The first is quite and the second is nearly independent of speed.

(2) The road resistance is expressed in pounds per ton, or preferably as a percentage of the vehicle weight ; the gradient resistance as 1 in G, hence as $\frac{W}{G}$; and the air resistance is expressed as the frontal area of the vehicle multiplied by the square of its velocity in feet per second multiplied by a suitable constant depending on the *shape* of the vehicle as a whole.

So the tractive resistance

$$T_R = W\left(R + \frac{2240}{G}\right) + kV^2A$$

when W = vehicle weight in *tons*
R = rolling resistance in pounds per ton
G = gradient
V = velocity in f.p.s.
A = the projected area of the vehicle

We have seen that the value of $W\left(R + \frac{2240}{G}\right)$ can at once be found from Table III or the diagrams (Figs. 4 and 5), and the values of kV^2A are given in Table IV and only require to be multiplied by the actual projected area of the vehicle. We will take two examples, first a touring car—

Total weight	W = 1·5 tons
Frontal area	A = 25 sq. ft.
Gradient	G = 1 in 7
Road resistance	R = 50 lb. per ton
Speed	V = 40 m.p.h. 58·6 f.p.s.

Then from the formula

$$\begin{aligned} T_R &= 1{\cdot}5\left(50 + \frac{2240}{7}\right) + (\cdot0017 \times 58{\cdot}6^2 \times 25) \\ &= 1{\cdot}5\,(370) + 146 \\ &= 555 + 146 = 701 \text{ lb.} \end{aligned}$$

Or working from the Table III we find the value of the road resistance of 50 lb. per ton to be 2·23 per cent and the gradient resistance of 1 in 7, 14·3 per cent

$$14{\cdot}3 + 2{\cdot}23 = 16{\cdot}53 \text{ per cent}$$

The gross weight of the vehicle is 1·5 tons, or 3,360 lb., and 16·53 per cent of this is $\frac{3360 \times 16{\cdot}53}{100} = 555$ lb. as before, which is the simple way.

FIG. 4. PERCENTAGE OF VEHICLE WEIGHT DUE TO ROAD RESISTANCE

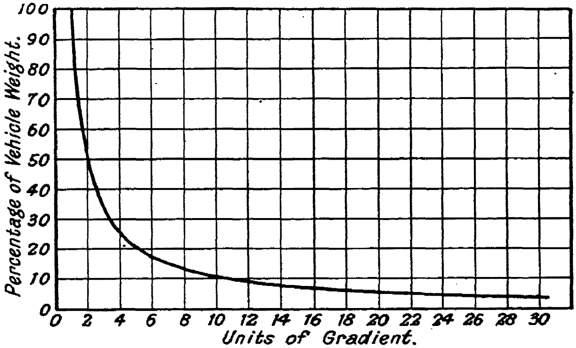


FIG. 5. PERCENTAGE OF VEHICLE WEIGHT DUE TO UNITS OF GRADIENTS

Next we take the case of a lorry weighing, when loaded, 7 tons, and this will be moving so slowly uphill

that in normal circumstances the air resistance may be neglected. We assume the following conditions—

Total weight	$W = 7$ tons
Gradient	$G = 1$ in 6
Road resistance	$R = 80$ lb. per ton

The speed and the frontal area are neglected. Then by formula

$$T_R = W\left(80 + \frac{2240}{6}\right)$$
$$= 7\ (453\cdot33) = 3173\cdot31 \text{ lb.}$$

And again working this by percentages

$$R = 3\cdot57 \text{ per cent}$$
$$G = \underline{16\cdot6} \quad ,,$$
$$\text{Total } 20\cdot17 \quad ,,$$

$$W = 7 \times 2240 = 15680 \text{ lb.}$$

and 20·17 per cent of this $= \dfrac{15680 \times 20\cdot17}{100} = 3173$ lb.

We now have to connect this with the tractive effort of the vehicle. This we know to be the actual torque at the road wheel, which is found from the formula—

$$T_E = \frac{T \times e \times r}{\dfrac{D}{2} \times \dfrac{1}{12}}$$

where T = engine torque in pounds feet

e = transmission efficiency = 80 per cent

r = total gear ratio (gear box × rear axle)

D = wheel diameter in inches = 36

then $\dfrac{D}{2} \times \dfrac{1}{12} = \dfrac{36}{24} = 1\cdot5$ ft.

We will assume an engine torque of 200 lb.-ft.

then $$\frac{200 \times \cdot 8 \times r}{1\cdot 5} \quad T$$

whence $$106\cdot 6 \times r = T_{\mathrm{E}}$$

and since T_{E} must equal T_{R}, and $T_{\mathrm{R}} = 3{,}173$ lb., we get

$$106\cdot 6r = 3173$$

whence $$r = \frac{3173}{106\cdot 6} = 29\cdot 75$$

which is therefore the required gear ratio.

So $$\frac{T \times e \times r}{\frac{D}{24}} = W\left(R + \frac{2240}{G}\right)$$

and in the case of a fast car the formula becomes

$$\frac{T \times e \times r}{\frac{D}{24}} = W\left(R + \frac{2240}{G}\right) + \cdot 0017\ V^2 A$$

and the value of any symbol can be obtained by equating from this.

A very important point about the torque curve of a petrol engine must be noticed here. If we turn to Fig. 3 we see that the torque curve cuts the 200 lb.-ft. line at *two* speeds, viz. at 400 and at 1,600 r.p.m.; at either of these speeds, therefore, the effort of the engine will exactly balance a resistance of 200 lb.-ft. It will not, however, be possible to run an engine at both of these speeds against such a resistance. Suppose first the engine is running at 1,600 r.p.m., it is certain that minute fluctuations of speed must occur; if the speed

falls slightly, the torque curve *rises* and the engine will overcome the resistance and the speed will be restored, but if the engine were running at 400 r.p.m. and the speed fell slightly, it is seen that the torque would be lessened and the engine, unable to recover itself, would rapidly slow down and finally stop. The speed of stable running is not exactly at the point of maximum torque but slightly higher; in the case of the engine referred to in Fig. 3, its stable speed for heavy pulling would be about 1,500 r.p.m.

We have next to deal with the speed of the vehicle; we know that the term horse-power is a convenient way of expressing the *rate* at which an engine performs work, and we are not at the moment concerned with the power developed at the engine, but the power which is available at the driving wheel. This power may be expressed as

$$\text{h.p.} = \frac{\text{tractive force in lb.} \times \text{ft. per min. speed}}{33000}$$

So for any one horse-power this tractive force and the vehicle speed would be inversely proportional, and a graph of the two would be a rectangular hyperbola.

For example: Let the tractive force = 625 lb. and the speed 30 m.p.h. or 2,640 ft. per min.

then $$\frac{625 \times 2640}{33000} = 50 \text{ b.h.p.}$$

So at 15 m.p.h. the tractive force would be 1,250 lb. and at 60 m.p.h. 312·5 lb. These figures are shown in Fig. 6, where pounds tractive effort are plotted on speeds in miles per hour.

If the internal combustion engine gave a steadily increasing torque with falling speed we could attain this ideal, but unfortunately it does not do so, and the

interposition of gearing becomes necessary in order to provide the maximum torque at any road speed. It is at once clear that an infinity of gear ratios would be required for such a purpose, and so far no such mechanism has been invented, but by providing a limited number of gears, and taking advantage of the flexibility

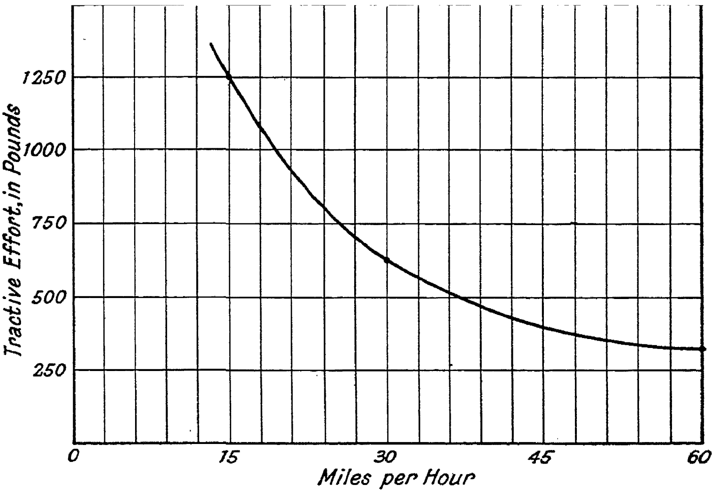


FIG. 6. CURVE OF IDEAL TRACTIVE EFFORT

which, to a greater or less extent, all petrol engines possess, it is possible to approximate to the ideal conditions. We remember there is an invariable gear in the back axle, which may vary from 3 to 1 up to 5 to 1 in a touring car, and from 5 to 1 up to, say, 10 to 1 in the case of a lorry or omnibus. We will take the case of a light lorry fitted with the engine whose performance is illustrated in Fig. 3, furnishing a torque of 200 lb.-ft. at 1,600 r.p.m. We assume this fitted in a lorry which, fully loaded, weighs, say, 8,000 lb., and

with the following specification for the rest of the vehicle—

Diameter of wheels, 36 in.
Back axle ratio, 5 to 1

Gear box ratios . .	4th speed,	1 to 1 =	total	5 to 1
	3rd ,,	1·817 to 1 =	,,	9·09 to 1
	2nd ,,	3·3 to 1 =	,,	16·5 to 1
	1st ,,	6 to 1 =	,,	30 to 1
Mechanical efficiency e on 1st, 2nd, and 3rd speeds				·8
,, ,, e on 4th speed				·9

We require to produce torque curves at the road wheels which correspond to that of the engine when operating on each of these gears, and as the wheel speed determines the speed of the vehicle in miles per hour, we will employ these units instead of r.p.m. Referring then to Table I, we find the torque corresponding to the speed of the engine, and if we multiply this by the respective gear ratios, and by the mechanical efficiency of each, we shall obtain the torque in pounds -feet at the wheels.

Dividing this by the wheel radius in feet we get the actual pounds tractive effort. Again, if we divide the engine speed by the gear ratios we obtain the r.p.m. of the road wheels, and we have to convert this into miles per hour. Thus, if

$$n = \text{r.p.m. of the engine}$$

$$60n = \text{r.p.h.} \quad ,, \quad ,,$$

and $60\dfrac{n}{r} = $ r.p.h. of the road wheels

where $r =$ the total gear ratio in use

1 mile $= 5{,}280$ ft. and the wheel circumference

$= \dfrac{\pi}{12} D$ ft. where D is the diameter in inches. Therefore

$$\text{revs. of wheel per mile} = \frac{5280}{\dfrac{\pi}{12} D}$$

and miles per hour of vehicle =

$$\overline{\frac{\pi}{12}D}$$

$$\text{or} = \frac{60 \times \cdot 26179}{5280} \times \frac{nD}{r} = \frac{0{\cdot}002975nD}{r}$$

and for a 36 in. wheel, miles per hour $= m = 0{\cdot}1071 \times \dfrac{n}{r}$

We then proceed to convert the figures of engine speed into wheel speeds at the four gear ratios, and these are next tabulated.

TABLE V

Engine r.p.m.	Vehicle speeds miles per hour for four gear ratios			
	1st, 30/1	2nd, 16·5/1	3rd, 9·09/1	4th, 5/1
500	1·785	3·245	5·89	10·7
1,000	3·570	6·490	11·78	21·4
1,500	5·355	9·735	17·67	32·1
2,000	7·140	12·980	23·56	42·8
2,500	8·925	16·225	29·45	53·5

Next we have to find the tractive effort at the corresponding speeds, and this is

$$T_E = \frac{T \times e \times r}{\frac{D}{24}} \quad \frac{24T \times e \times r}{D}$$

and as $D = 36$ in. $\qquad T_E = T \times e \times r$

TABLE VI

TRACTIVE EFFORT AT WHEELS FOR FOUR GEAR RATIOS (POUNDS)

Engine r.p.m.	1st, $r = 30$, $e = \cdot 8$	2nd, $r = 16\cdot 5$, $e = \cdot 8$	3rd, $r = 9\cdot 09$, $e = \cdot 8$	4th, $r = 5$, $e = \cdot 9$
500	3,248	1,788	984	609
1,000	3,340	1,840	1,014	627
1,500	3,250	1,796	989	612
2,000	2,975	1,637	902	558
2,500	2,590	1,427	786	486

The next operation is to prepare a tractive effort diagram and plot the torque curves for each gear ratio, as given in Table VI, on speeds in miles per hour from Table V, the latter ranging from 1·78 to 53·5 m.p.h. The result is seen in Fig. 7, where each of the four curves represents the actual tractive effort at each gear ratio. Then, through the third point of each curve which corresponds to the selected engine speed of 1,500 r.p.m. (selected as giving the best working point on the engine torque curve) we draw a dotted curve, the hyperbola of constant power, and this is seen to have the same characteristics as the ideal tractive effort curve of Fig. 6. Moreover, there is no part of this curve which is not overlapped by one or other of the speed ratio curves, and we find we have a suitable gear ratio for providing the maximum pulling power at any vehicle speed.

Now examine for a moment the four gear ratios of the gear box. These are $\frac{1}{1}$, $\frac{1\cdot 817}{1}$, $\frac{3\cdot 3}{1}$, and $\frac{6}{1}$. It is seen that each is 1·817 times the ratio of the previous one, hence they are in geometrical progression. In selecting a set of ratios this series should, as nearly as possible, be adhered to for easy gear changing, and when the

top gear has been determined upon, and the bottom to meet the worst conditions of tractive resistance the

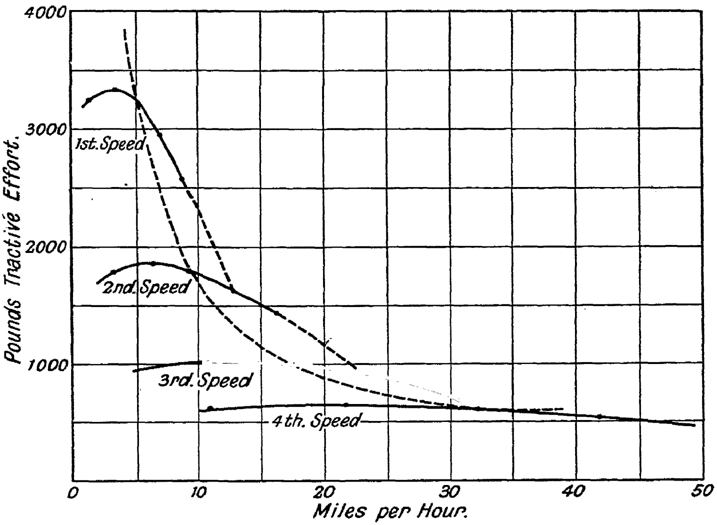


FIG. 7

vehicle is expected to encounter, the common ratios for the intermediate gears can be found thus,

$$d = \sqrt[3]{\frac{\text{bottom gear ratio}}{\text{top gear ratio}}} \text{ for four gears}$$

$$\text{and } d = \sqrt[2]{\frac{\text{bottom gear ratio}}{\text{top gear ratio}}} \text{ for three gears}$$

To conclude, therefore, it will be seen that for the propulsion of a motor vehicle it is not enough merely to settle the amount of engine torque that is needed, but in addition to see that the gear ratios from engine to road wheels are such as to enable the engine to run (at

any vehicle speed) at the specific speed at which the desired torque is available.

When a vehicle is ascending hills, the speed always falls back automatically to its best climbing speed, and if that is inadequate, the engine will stop unless the gear be changed. We will now see what can be learnt

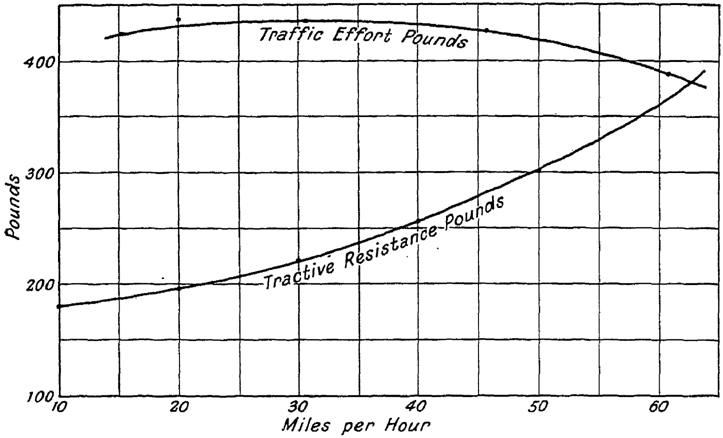


FIG. 8

from the engine torque and tractive resistance curves taken together.

Let us assume a touring car weighing, when fully loaded, 3,500 lb. travelling over a road which is approximately level but with occasional rises of about 1 in 30 having a good surface with a road resistance of about 40 lb. per ton. From Table III we find these resistances together will amount to, say, 5 per cent of the car's total weight, and this will be $\frac{3500 \times 5}{100} = 175$ lb. due to deviations from level and to rolling resistance. We assume, further, that the car has a frontal area of

30 sq. ft. and a wind pressure factor of ·00079, we find that at

	10	20	30	40	50	60 m.p.h.
the pressure is	·17	·68	1·53	2·72	4·25 and	6·12 lb. sq. ft.

Multiplying these by 30 we get a wind pressure of

5·1 lb. 20·4 lb. 45·9 lb. 81·6 lb. 127·5 lb. and 184 lb.

To each of these must be added 175 for road resistance, and the figures for total resistance then become

180 195 220 257 302 and 359 lb.

We proceed to plot this curve of tractive resistance against miles per hour as in Fig. 8.

We will make the gear ratio on fourth speed $\frac{3{\cdot}5}{1}$ and the wheels 36 in. diameter. Then the tractive effort will be

$$T_E = \frac{T \times {\cdot}9 \times 3{\cdot}5}{\frac{D}{24}} = 2{\cdot}1T$$

and from the engine torque curve of Table I we get a tractive effort at

	500	1,000	1,500	2,000 r.p.m.
of	426 lb.	439 lb.	428 lb.	390 lb.

The road speeds corresponding to these engine speeds will be as before $\cdot 1071 \times \frac{n}{r}$, namely—

15·3 30·6 45·9 61 m.p.h.

Plotting these values as in Fig. 8 we see that at 30 m.p.h. there is a considerable surplus of power, which rapidly decreases, until at about 62 m.p.h. the curves cross and the margin has disappeared.

It is evident that under the assumed conditions it will be physically impossible to travel faster than 62 m.p.h., and as a study of the figures show, the air resistance is largely responsible for this result.

The case is different with a lorry ascending a steep hill on a low gear. There is then practically no wind resistance, and the curve of tractive resistance is a straight horizontal line, and, provided a suitable gear ratio has been provided, the vehicle can keep on ascending for an indefinite period.

Referring again to Fig. 8 we see that at 30 m.p.h. the tractive resistance is only about one-half the tractive effort, which means that one-half of the latter at 30 m.p.h. is available for acceleration alone. In the case of passenger and racing cars acceleration is a very important factor, and this, of course, depends on the surplus power available after overcoming the gradient, surface, and wind resistance which the car encounters.

It is from analysis of this kind that we are able to select suitable engines for any desired road performance. Professor Reidler first analysed the various losses which occur in certain parts of a motor vehicle, such as rolling resistance, transmission losses, wheel slip, tyre losses, etc., and other investigators have extended his experiments. The first thing we have to do is to show all the characteristics of the engine at miles per hour instead of revolutions per minute, which can readily be done from the formula $m = \frac{\cdot 002975nD}{r}$ and we can thereafter assume any road conditions necessary. Let us take a further example and see how we can predict the road performance obtainable from either of two engines to be tested in the same vehicle.

We will take the same car as before, weighing 3,500 lb., and we will have two alternative engines having the power characteristics shown in Fig. 9. It

is seen that at 3,000 r.p.m. both engines deliver the same horse-power, but while one engine (*a*) is rather a

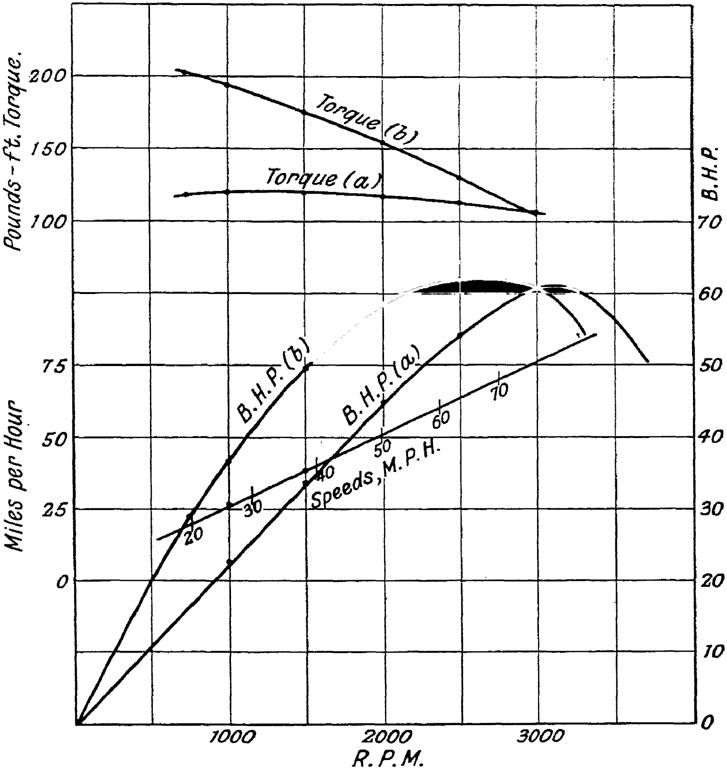


FIG. 9

thoroughbred type, whose power curve rises in a straight line, the other is a less refined design whose power curve is very different.

The wheel diameter is 34 in. and the top gear ratio is 4 to 1, so we translate the engine speeds in revolutions

per minute into corresponding speeds in miles per hour, using the formula $m = \dfrac{\cdot002975Dn}{r} = \cdot0253n$

R.P.M.	750	1,000	1,500	2,000	2,500	3,000
M.P.H.	19	25·3	38	50·1	63·2	76

TABLE VII

HORSE-POWER AND TORQUE OF ENGINES. (See Fig.

	Engine (*a*)		Engine (*b*)	
R.P.M.	B.H.P.	Torque lb. ft.	B.H.P.	Torque lb. ft.
750	17	119	29	203
1,000	23	120·7	37	194
1,500	34	119	50	175
2,000	45	118	59	155
2,500	54	113·5	62	130
3,000	61	107	61	107

By plotting these figures on Fig. 9 we can obtain the torque values at intermediate speeds, thus—

M.P.H.	Torque Engine (*a*)	Torque Engine (*b*)
20	119	205
30	120	190
40	120	170
50	119	155
60	115	135
70	110	115

The corresponding torque at the wheels will be

$$T_E = \frac{T \times \cdot 9 \times 4}{\frac{D}{24}} = \frac{24T \times \cdot 9 \times 4}{36} = 2{\cdot}4T = T_E$$

The tractive effort therefore becomes—

M.P.H.	T_E (a)	T_E (b)
20	286	492
30	288	456
40	288	408
50	286	372
60	276	324
70	264	276

We assume the same frontal area as before, viz., 30 sq. ft., giving wind pressures as follows—

M.P.H.	Wind Pressure	Wind Pressure and Other Variable Losses
	Lb.	Lb.
20	20·4	21
30	45·9	48
40	81·6	90
50	127·5	140
60	183·5	202
70	250	275

To these we must add something for windage losses in the wheels and friction of the tyres (hysteresis), which will increase from zero at 10 m.p.h. to 5 per cent at 30 m.p.h. and 10 per cent at 70 m.p.h., which will give the figures in the third column, and to each of these we must add the road resistance of, say, 45 lb. per ton or,

say, 2 per cent of the car's weight $\frac{3500 \times 2}{100} = 70$ lb., so the final figures of tractive resistance become

M.P.H.	20	30	40	50	60	70
T_R	91	118	160	210	272	345 lb.

Then by plotting the tractive resistance curve and the two tractive effort curves against miles per hour, as in Fig. 10, we see that the two tractive effort curves fall below the tractive resistance curve at 62 m.p.h. in the case of engine (*a*) and at 65 m.p.h. in the case of engine (*b*), and under the conditions assumed these are the maximum speeds possible with these engines. As we might have expected, therefore, from an inspection of Fig. 9, the attainable speeds for the car with either engine will be practically the same, but at lower ranges of speed the "excess power" of the (*b*) engine is approximately double that of the (*a*) engine. This excess power is, of course, available for hill climbing or for acceleration, and with this large reserve it is easy to see that while there is little, if any, choice between the engines for speed purposes, for comfortable driving the (*b*) engine is greatly superior.

Assuming the car is running on the level, *all* the excess power can be utilized for acceleration, and we will examine this next.

We start with the well-known formula $P = mf$, where P is the accelerating force, m the mass of the car $\frac{W}{32 \cdot 2}$ and f the acceleration in feet per second per second.

So $$P = \frac{W}{32 \cdot 2} \times f = \frac{3500}{32 \cdot 2} \times f = 108 \cdot 7f$$

Whence $f = \frac{P}{108 \cdot 7}$. Now P is the excess tractive effort available at the wheels at the various speeds, and

deducting the tractive resistance from the tractive effort we get for the two engines the following net values for the excess tractive effort in pounds—

Speed, m.p.h.	20	30	40	50	60
Engine (*a*)	195	170	128	76	4
Engine (*b*)	401	338	248	162	52

Using these values for P we get acceleration as follows—

M.P.H.	Acceleration, ft. per sec. per sec.	
20	Engine (*a*) 1·8	Engine (*b*) 3·7
30	,, 1·56	,, 3·1
40	,, 1·18	,, 2·28
50	,, ·7	,, 1·49
60	,, 0	,, ·48

From these figures, which are also plotted on Fig. 10, we see that with an engine of type (*a*) the car would be on top gear, a very sluggish performer, although with a four speed gear box handled by a skilful operator the result would be much better.

We have throughout the above investigation employed engine *torque*, but as many writers employ horse-power for the same purpose we will give an example of this as an alternative method.

Tractive effort at miles per hour multiplied by 88 gives tractive effort at feet per minute, and if this be multiplied by tractive resistance in pounds and divided by 33,000 the answer will be horse-power, or

$$\frac{m \times 88 \times P_R}{33{,}000} = \text{h.p.}$$

and by this we find the equivalent horse-power of tractive resistance for the various speeds, viz., 4·8, 9·9, 17, 28, 43·5, 64·4 b.h.p.

In Fig. 11 these values are plotted, and over them the b.h.p. of engine (*a*), Fig. 9, at the same speed

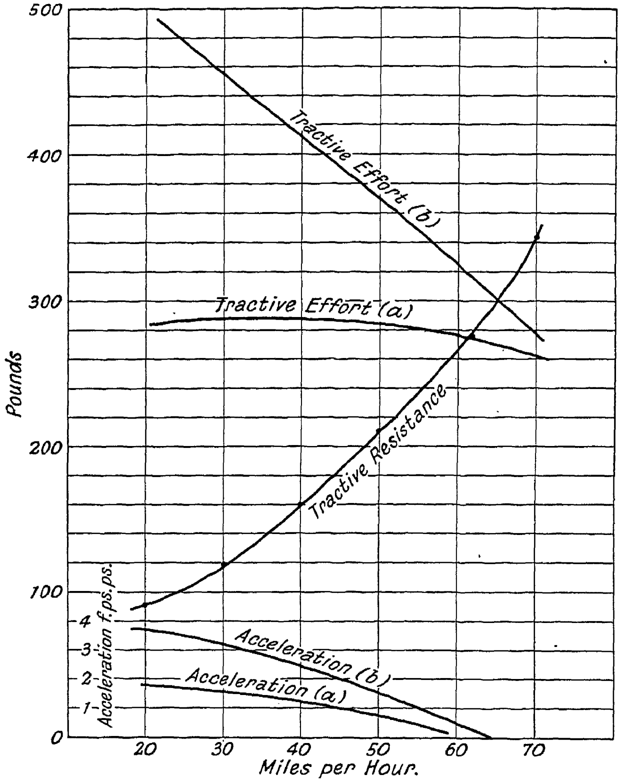


FIG. 10

units and the space between the two curves shows the excess horse-power available.

This method is not of such general use, as the figures of excess *horse-power* cannot be directly used for such calculations as are required for acceleration or gradient

resistance, and the use of the torque values furnishes figures more readily employable for these purposes.

Let us again turn to the chart in Fig. 7 and assume the data there given to be applied to a lorry which,

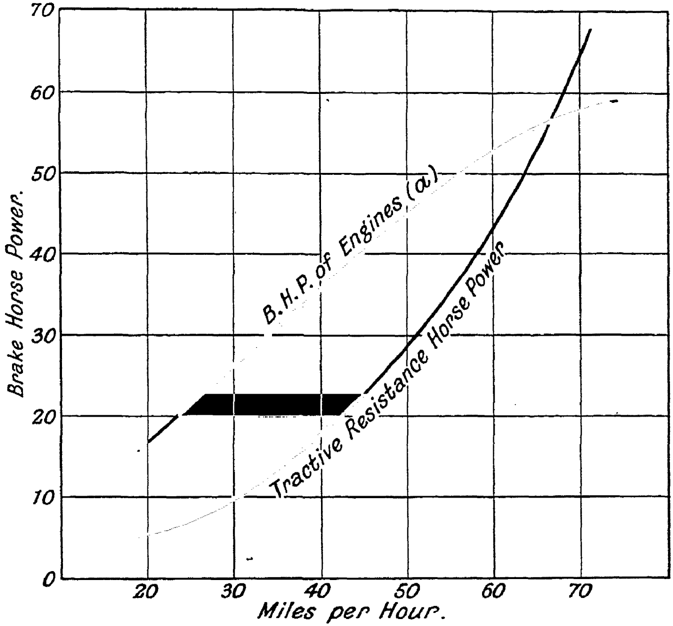


FIG. 11

when fully loaded, weighs 8,000 lb., the engine power, gear ratios, and wheel diameter all as before. We neglect the question of air resistance as when running on gears the speeds will not produce very material pressure. We must, however, have some figure for rolling resistance, and we will assume between 50 and 60 lb. per ton, or, say, 2·5 per cent of the vehicle weight; this amounts to 200 lb., and all the tractive effort over and

above this will be excess power for ascending gradients. From the diagram we see that the best pulling speed on each gear is—

32 m.p.h.	on	4th speed	with a	tractive	effort of	600 lb.
18	,,	3rd	,,	,,	,,	1,000 ,,
9·5	,,	2nd	,,	,,	,,	1,800 ,,
5·5	,,	1st	,,	,,	,,	3,250 ,,

and deducting 200 lb. for rolling resistance from each of these we are left with 400 lb., 800 lb., 1,600 lb., 3,050 lb. excess power on each speed, and since

$$P = \frac{W}{G},\ G = \frac{W}{P} = \frac{8000}{\text{excess power}}$$

Substituting the figures we have just found, we get

on 4th speed, $G = \frac{8000}{400} = 1 \text{ in } 20 =$ a gradient of 5 per cent

,, 3rd ,, $G = \frac{8000}{800} = 1 \text{ in } 10 =$,, 10 ,,

,, 2nd ,, $G = \frac{8000}{1600} = 1 \text{ in } 5 =$,, 20 ,,

,, 1st ,, $G = \frac{8000}{3050} = 1 \text{ in } 2{\cdot}6 =$,, (approx.) 40 ,,

In Fig. 12 the tractive effort curve is plotted on miles per hour with a quadrant described from the zero point, and through each of the selected points a radius is drawn. Where the radii cut the circle we have a scale of gradients starting from the base of the diagram, and the intermediate points would show the climbable gradients at any point of the tractive effort curve. Again, from Fig. 7, we see that at

13 m.p.h.	we can use	1st or 2nd	speed
23 ,,	,,	2nd or 3rd	,,
32 ,,	,,	3rd or 4th	,,

and gradients corresponding to radii drawn through these points could be ascended on either of two gears,

though for stable running the lower of the two would be preferable.

We saw that the tractive factor T_F is the ratio of tractive effort to total weight, or $\frac{T_E}{W}$, both quantities

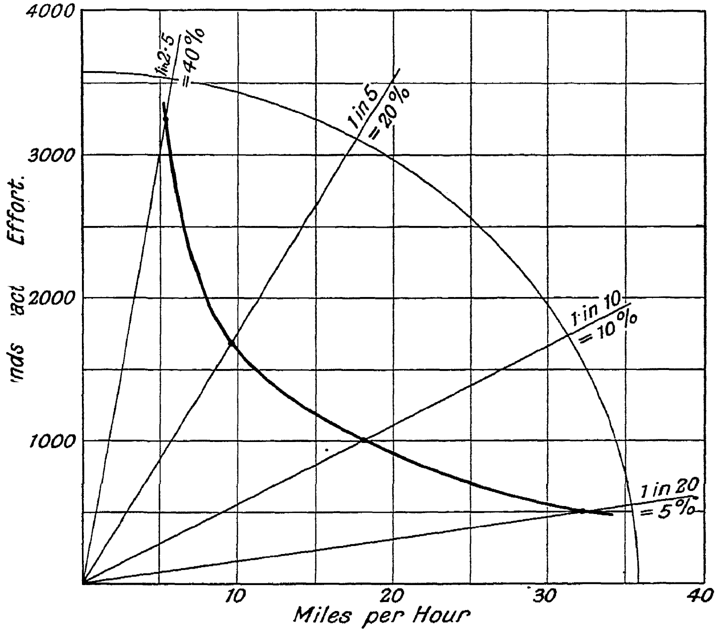


FIG. 12

being in pounds. We can write the equation for acceleration

$$T_E = mf \text{ or } T_E = \frac{W}{32{\cdot}2} \times f, \text{ whence } f = \frac{32{\cdot}2 T_E}{W}$$

and substituting the value of $T_F = \frac{T_E}{W}$ it follows that $f = 32{\cdot}2 T_F$ ft. per sec. per sec.

Let us take a numerical example—

Let $T_E = 500$ lb. and $W = 3{,}000$ lb.

then $\dfrac{T_E}{W} = \dfrac{500}{3000} = 0{\cdot}166$

then $f = 32.2 \times 0{\cdot}166 = 5{\cdot}37$ ft. per sec. per sec.

As the tractive factor is easily determined, and must in fact enter largely into the design of any vehicle, it follows that this is by far the simplest way of calculating the acceleration; we shall now learn what is the limiting factor to this. A little thought will show that the tractive force we can put into the wheels must depend upon the point at which the wheels themselves will slip on the road surface.

Adhesion. We come then to the consideration of another factor, that of road adhesion, or the force with which friction enables the wheel to cling to the surface of the road; this is usually expressed by the symbol μ, and it enters into all calculations relating to driving and braking.

It is not an easy matter to determine the exact coefficient of friction between the tyre and the road, but many investigators have contributed to the knowledge available, and while it has long been customary to assume the value of μ to be 0·6 it has been established that much higher values may be expected. Laboratory experiments, for instance, show that for pneumatic car tyres with a suitable tread, the coefficient of friction, when running on steel, wood, and glass, is respectively 1·72, 1·69, and 1·93. This fact is, to say the least, unexpected, and experiments on actual roads give lower figures, but it is not unusual to observe values as high as 0·8 or even 0·9, but it is not possible to state an empirical figure since the road surface is such a variable quantity. We may, however, with modern low pressure

tyres with good treads, safely take the figure as being anything between 0·6 and 0·75 on good dry roads, such as concrete or hard tar macadam. On the other hand, with wet and greasy roads, and more particularly under wet conditions, with certain kinds of asphalt the value is very much less and under skidding conditions it nearly disappears. With 0·7 as the value we mean that if the load on one wheel is 1,000 lb., that wheel can exert a driving force of 700 lb. (whether driving or braking) without slipping.

In the case of our lorry we know that the maximum tractive effort on first speed is 3,340 lb. (distributed between two wheels); the vehicle weighs 8,000 lb., of which, under full load conditions, perhaps 75 per cent is carried on the rear axle. The two wheels are therefore pressing on the road with a weight of 6,000 lb., and this, multiplied by 0·7 (the value of μ), gives 4,200 lb. as the maximum force of adhesion; so we see that with the gear ratios and engine power we have chosen, the wheels would not spin even when the first gear was used with the full torque of the engine. We thus see another aspect of the matter, that in designing any vehicle we can start from the adhesive force possible, making a liberal allowance for overloading and also for a high coefficient of road friction, and say, that is the maximum tractive effort we can ever encounter irrespective of the road surface or gradient, and choose our first speed ratio accordingly, using this and the top speed from which to establish the intermediate ratios.

In a touring car it almost invariably happens that on first speed the wheels can be made to spin under certain conditions.

It follows that the question of adhesion has a material influence on the possible acceleration, for if $\mu =$ the coefficient of friction or adhesive factor for the driving

wheels, and w is the weight on the driving axle, W being the total weight as before, it is evident that

$$T_E = w\mu$$

hence by substitution $w\mu = \dfrac{Wf}{32{\cdot}2}$

whence $$f = \frac{32{\cdot}2 w\mu}{W}$$

and this is the maximum acceleration attainable without skidding or slipping the wheels.

There are vehicles made for special purposes where the drive is transmitted through all four wheels, and in these it is seen that w, the weight on the driving wheels is the same as W. In this case the acceleration is $f = \dfrac{W\mu g}{W} = \mu g$ as the weights have cancelled out, and as we have seen it is possible under certain conditions to find the value of μ to be actually equal to or greater than 1, the acceleration under these conditions would amount to $f = g = 32{\cdot}2$ ft. per sec. per sec. Such a rate of acceleration would be, however, quite insupportable, the average person finding a rate of 8 ft. per sec. per sec. as high as can comfortably be borne.

There is another influence which adversely affects acceleration, and it is the inertia of the wheels and other rotating parts of the chassis which together offer considerable resistance. With the modern types of wheel, however, this is reduced very considerably as compared with the wooden wheels formerly used. On the other hand, the present tendency is to employ large and relatively heavy pneumatic tyres, which have a considerable flywheel effect. This may amount to about 5 per cent of the total tractive resistance when starting from rest on the level, but, as a rule, this factor

may be neglected for any but academic purposes, and with all the factors taken into account there are so many causes of inaccurate data that at the best calculations must be recognized as being of only an approximate nature.

There are certain effects of running on a gradient which we have not yet noticed. We saw at the commencement that the force P required to propel a vehicle *uphill* could be calculated by the ordinary equation for the forces on an inclined plane. Similarly if the vehicle is running *downhill*, the equation, using the same notation, becomes

$$P = \frac{W \sin (\varphi - \alpha)}{\cos \varphi}$$

We will take a numerical example of a vehicle on a hill first travelling uphill on a gradient of 1 in 3 using the formula $P = \dfrac{W \sin (\alpha + \varphi)}{\cos \varphi}$

$$1 \text{ in } 3 = \frac{h}{l} = \frac{1}{3} = \cdot 333 = \sin \alpha \text{; so } \alpha = 19^\circ\ 27'$$

let $R = 50$ lb. per ton $= c$ the coefficient of road resistance

then $\log c = \log \tan \varphi$

$$50 \text{ lb. per ton} = \frac{50}{2240} = \cdot 0223 \text{ and } \log \cdot 0223 = \bar{2}\cdot 3488$$

this equals $\log \tan \varphi$ whence $\varphi = 1^\circ\ 16'$

$$\text{then } P = \frac{W \sin (\alpha + \varphi)}{\cos \varphi} = W \frac{\sin (19^\circ\ 27' + 1^\circ\ 16')}{\cos 1^\circ\ 16'}$$

$$\therefore P = W \frac{\sin 20^\circ\ 43'}{\cos 1^\circ\ 16'} = W \frac{\cdot 3537}{\cdot 9997} = W \times \cdot 354$$

Let $W = 1{,}000$ lb., then $P = 1{,}000 \times \cdot 354 = 354$ lb.; or, by the simpler method,

$$\frac{W}{3} = 333, \; R = \frac{50 \times 1000}{2240} = 22{\cdot}33 \text{ lb.}$$

$$333 + 22{\cdot}33 = 355 \text{ lb.}$$

Again, if the car weighs 3,000 lb.

$$R = \frac{50 \times 3000}{2240} = 67 \text{ lb.}$$

Then ascending $P = \dfrac{3000}{3} + 67 = 1{,}067$ lb.

and descending $P = \dfrac{3000}{3} - 67 = 933$ lb.

When descending the acceleration due to the gradient alone can be found from the formula $f = \dfrac{32{\cdot}2P}{W}$

$$\text{So } f = \frac{32{\cdot}2 \times 933}{3000} = 10{\cdot}37 \text{ ft. per sec. per sec.}$$

The gradient, if at all steep, has another effect, namely that of altering the load distribution with important changes in the conditions. As the centre of gravity is always above the wheel centres, it follows that in ascending more weight is thrown on the rear axle, and conversely in descending more on the front axle, the proportion thus transferred depending on the height of the centre of gravity and its horizontal position relative to the two wheels. The full discussion of this belongs to another place, but it is appropriate to mention it here as it affects driving and tractive effort as well as brakes.

If a car is on the level, the weight on each axle will be, if W is the total weight, b is the wheel base, and l_R

and l_F the distance of the centre of gravity from the axles R and F respectively (see Fig. 13).

Then under static conditions the weight at $F = (W_F)$ is $W_F = W \times \frac{l_R}{b}$, and at R, $W_R = W \times \frac{l_F}{b}$, but when standing on a gradient the weight tends to be thrown more and more on to the *lower* wheel as the gradient is increased, and this effect is the same whether the car is ascending or descending. In the former case this change is advantageous as it gives a better hold for the

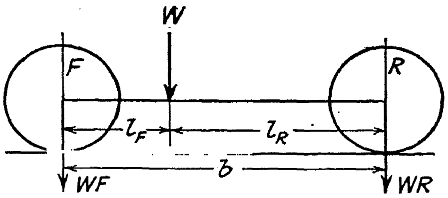


FIG. 13

driving wheels, but in descending, unless the vehicle has front wheel brakes, a large proportion of the holding power of the rear wheels is lost.

Fig. 14 shows a car, diagrammatically, on a gradient; the position of the centre of gravity CG determines the result; the weight on the wheels is shown as w_1 and w_2,

$$\text{then } w_1 = \frac{W\ (x \cos\theta + H \sin\theta)}{B \cos\theta} = \frac{W\ (x + H \sin\theta)}{B}$$

and of course $w_2 = W - w_1$.

The position of the centre of gravity longitudinally can be found by taking the weights on each axle, and by the equation of moments finding values for l_F and l_R (Fig. 13).

The height of the centre of gravity can only be determined by tilting the vehicle sideways on two wheels,

until it is just in a state of balance; the point of intersection of a line, projected vertically from the centre of the load bearing tyre tread, with the centre line of the vehicle will give the centre of gravity.

The height of the centre of gravity affects many problems of suspension, particularly the stability of the vehicle when rounding curves, since the centrifugal force tending to upset it acts through that point, and

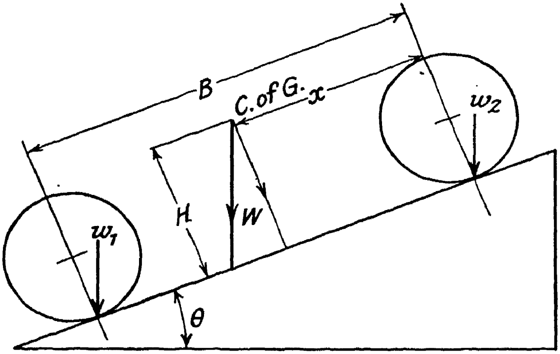


FIG. 14

when the couple of the force becomes equal to that of the weight on the *inner* wheel when rounding the curve the car will commence to run on two wheels; if conditions are maintained, the car will tilt until the perpendicular from the centre of gravity falls outside the track of the wheels, when the car will overturn.

Let us assume a lorry with a high load, as shown in Fig. 15, the weight W we assume 3,000 lb., then $\frac{W}{2}$ $= 1{,}500$ lb. $B = 5$ ft. and $h = 3$ ft. Then the force P is

$$P \times \frac{h}{B} = \frac{W}{2}, \text{ or } P\,\frac{3}{5} = 1{,}500, \text{ so } P = 2{,}500 \text{ lb.}$$

but P is the centrifugal force of the vehicle on a curve.

Now $CF = \frac{Wv^2}{gr}$; where v = velocity in feet per second, r = radius in feet, and $g = 32{\cdot}2$, let us assume a radius of 15 ft., and we require to know the overturning speed,

or $6{\cdot}21v^2 = 2{,}500$, so $v^2 = 400$ and $v = 20$ ft. per sec. $20 \times 1{\cdot}467 = 29{\cdot}34$ m.p.h., so beyond that on the

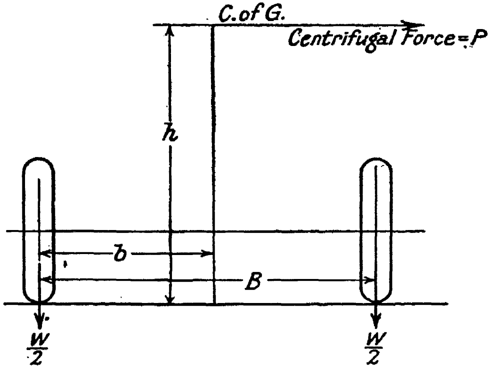


FIG. 15

radius assumed the vehicle will turn over. If we examine these equations we shall learn another interesting fact, thus

$$P \times \frac{h}{B} = \frac{W}{2}, \text{ so } \frac{2Ph}{B} = W \text{ and } P = \frac{WB}{2h}$$

but the centrifugal force when overturning is P.

So $CF = \frac{Wv^2}{gr} = \frac{WB}{2h}$ $\qquad$ W's cancel out.

So $\frac{v^2}{gr} = \frac{B}{2h}$ and $v^2 = \frac{32{\cdot}2rB}{2h}$

Substituting known values, $v^2 = \frac{32{\cdot}2 \times 15 \times 5}{2 \times 3} = 402$ ft. per sec. as before.

Hence we see that the *dimensions* of a car and the position of the centre of gravity determine its safety on curves, and providing these conditions remain constant, the weight has nothing to do with the matter. The same rather unexpected fact occurs also in braking problems which are dealt with elsewhere, and where the weight of the vehicle does not affect the stopping distance, since the energy of the moving vehicle is a function of W and so also is the stopping force, and the two cancel out.

This brings us to the question of the energy in a car. A body is said to possess energy when it can perform work, and it may possess this property through (*a*) its position, or (*b*) its condition.

(*a*) If the body is so held that if released it can perform work it is said to possess *potential energy*. Thus, a car standing on a gradient has potential energy equal to its weight multiplied by the *vertical* height through which it will fall when descending the gradient, since the work it will perform in so doing is $W \times h$ ft.-lb. So

$$\text{Potential energy} = E_p = W \times h \text{ ft.-lb.}$$

(*b*) The energy due to its condition is *kinetic energy*, or energy of motion; this is a function of its speed and its mass, and it follows that the kinetic energy of a moving car may be the result of its potential energy. The kinetic energy is found from the following equation

$$\text{Kinetic energy} = E_k = \frac{Wv^2}{2g} \text{ ft.-lb.}$$

where W is the weight of the moving body in pounds and v is its velocity in feet per second.

So, to take an example, if a car weighing 3,000 lb. is

travelling at a speed of 30 m.p.h., or 44 ft. per sec., its kinetic energy will be

$$E_k = \frac{3000 \times 44^2}{64 \cdot 4} = 90,180 \text{ ft.-lb.}$$

From this has to be deducted the rolling resistance which, as we know, is an insignificant amount in comparison. As the known speed of a car is generally stated in miles per hour, a more convenient form for the equation is

$$E_k = 0 \cdot 0334\, W V^2$$

where V is in miles per hour. In the case of brake calculations, therefore, the kinetic and the potential energy have to be absorbed by the brakes and there converted into heat before the vehicle can be stopped. This is more fully dealt with in the section on brakes, so we will only take one example. A vehicle weighing 8,000 lb. is travelling down a gradient of 1 in 7, which has a road resistance of 60 lb. per ton, at a speed of 20 m.p.h. and is stopped in 42 ft. What energy has to be absorbed by the brakes?

First, dealing with its potential energy, in 42 ft. the vehicle will have fallen through $\frac{42}{7} = 6$ ft.,

$$E_p = W \times h = 8000 \times 6 = 48,000 \text{ ft.-lb.}$$

Its kinetic energy at 20 m.p.h. is

$$E_k = \cdot 0334 \times 8000 \times 20^2 = 107,000 \text{ ft.-lb.}$$

The road resistance is $\frac{60 \times 8000}{2240} = 214$ lb., and this has been operating through a distance of 42 ft., $214 \times 42 = 9,000$ ft.-lb. (approx.).

The total energy therefore is

$$48,000 + 107,000 - 9000 = 146,000 \text{ ft.-lb.}$$

and the brakes will have to convert this into heat during the time the vehicle is travelling 42 ft.

The kinetic energy of a vehicle is a very serious factor in all the phenomena of skidding, and it explains why a skid may sometimes occur with such velocity, and with frequently such disastrous results. It will be seen that if, by reason of a very greasy surface, the holding power of the tyres is suddenly withdrawn, the kinetic energy stored in the vehicle becomes available for the most disastrous movements, which are often seemingly quite disproportionate to the speed at which the car may be travelling at the time.

For example, if the car is travelling at a moderate speed on a curve, and the rear wheels suddenly commence to skid, the whole of the kinetic energy of the vehicle acting through its centre of gravity swings the rear portion of the car round the front wheels as a fulcrum with somewhat unexpected results. The phenomena of skidding, however, are not easy to analyse, and it is not necessary to pursue the matter farther.

We have now covered all the important mechanical problems which affect a motor vehicle from without. The resulting requirements of the details of the mechanism of the chassis are dealt with in their appropriate places.

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